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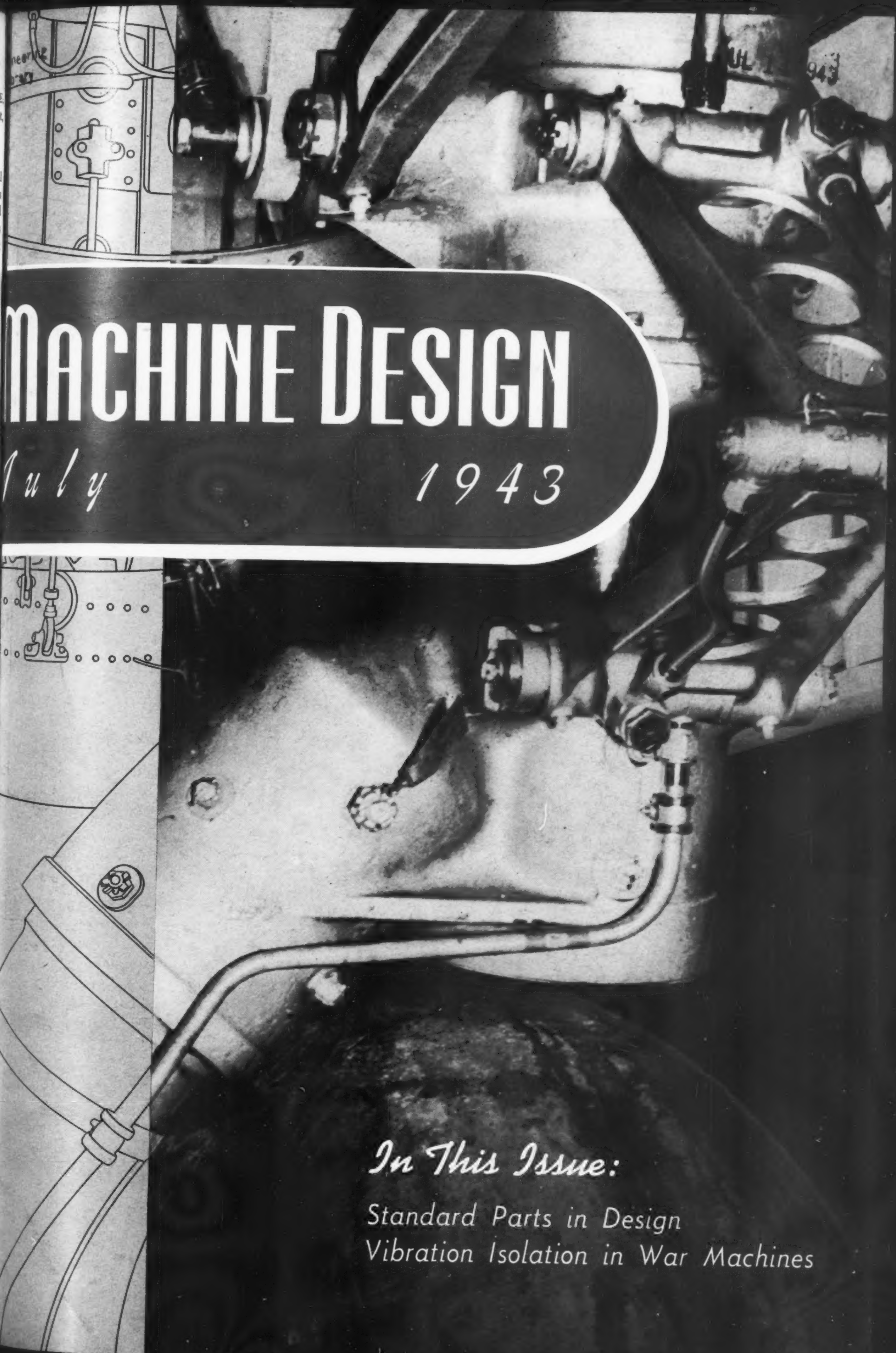

MACHINE DESIGN

THE PROFESSIONAL JOURNAL OF CHIEF ENGINEERS AND DESIGNERS

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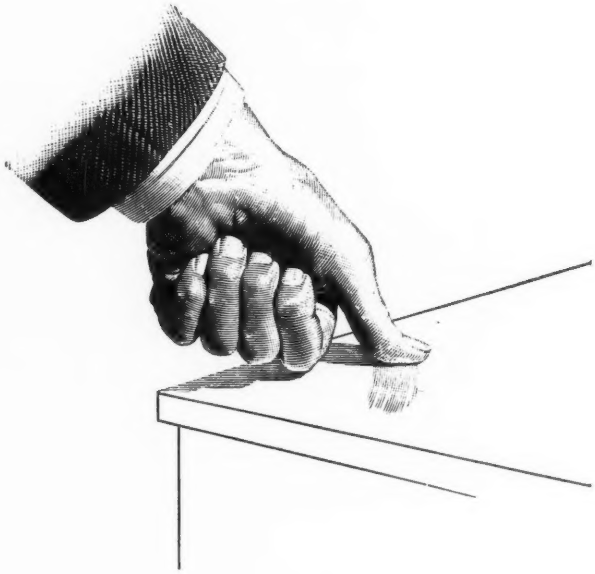
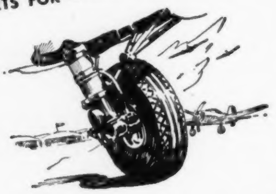


Itemized Index

Classified for Convenience when Studying Specific Design Problems

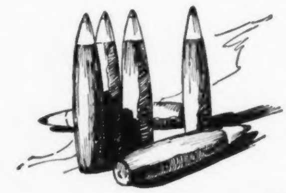
MACHINE DESIGN is indexed in Industrial Arts Index and Engineering Index Service, both available in libraries generally.

PARTS FOR — AIRCRAFT



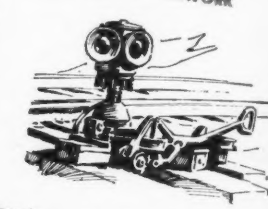
Tail wheel assemblies, actuating cylinders and engine forgings. Brake lining. Camshaft and starter bearings.

PARTS FOR — BATTLE



Shell and shot forgings. Cast bomb parts. Steel armor plate castings. Ammunition hoists. Torpedo parts.

PARTS FOR — TRACKWORK



Crossings, switch stands, switch point locks, guard rails, rail lubricators. Trackwork for industrial equipment.

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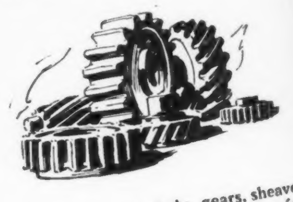

Manganese steel chain, gears, sheaves, sprockets and wheels. Hard surfacing welding rods. Forgings, castings.

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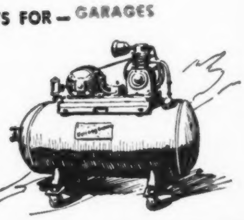
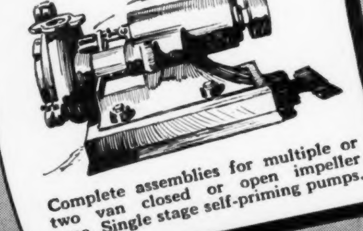


Air compressors. Car washers. Brake lining, fan belts, clutch facing, radiator hose, auto lifts.

INDUSTRIAL PUMPS



Complete assemblies for multiple or two van closed or open impeller type. Single stage self-priming pumps.

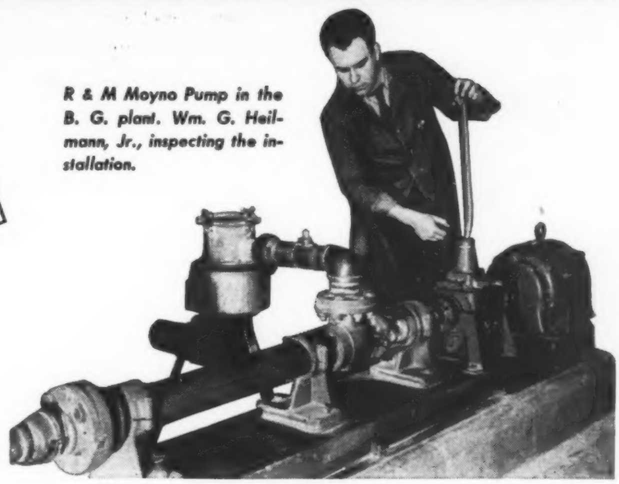
PARTS FOR — WELDING

Welding rods, flux, nickel-manganese filler, applicator bars. Welding products for repair of ferrous equipment.

PARTS FOR — SHIPS

Strut and stern tube bearings. Carbon steel castings for davits and winches. Bronze castings for engines.

That white substance in the hand above is ceramic slip—a material so abrasive that it quickly destroys the efficiency of conventional pumps! Yet one pump—the R & M Moyno—handles it with the greatest of ease. In the Ridgefield, N. J., plant of The B. G. Corporation, for example, a Moyno has been moving great quantities of ceramic slip for approximately a year with absolutely no maintenance except routine lubrication!



R & M Moyno Pump in the B. G. plant. Wm. G. Heilmann, Jr., inspecting the installation.

The list of things that an R & M Moyno can successfully pump is amazing. Such materials as molten resin, vitreous enamel, glucose, gun-boring coolants, propane gas, liquid carbon dioxide—the Moyno takes them in its stride. Virtually *any* abrasive or *any* heavy viscous liquid that can flow through a pipe, is moved faster and more economically with the Moyno.

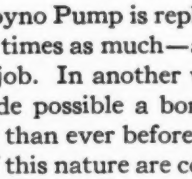
In one war industry a single Moyno Pump is replacing two other pumps that cost ten times as much—and the Moyno is doing a far better job. In another war plant a Moyno has made possible a boring speed many times faster than ever before attained. Performances of this nature are common in plants where Moyno Pumps are used.

There is no piston or valve wear with a Moyno—because Moyno has no pistons or valves. Cavitation and pulsation are eliminated—because the exclusive Moyno principle produces a flow that is constant and uniform. And the compact design is so simple that an installation can be completely dismantled and reassembled in 30 minutes.

If you have a pumping problem, don't fail to get the full facts about the R & M Moyno. Write us, stating your particular problem, and we will tell you how the Moyno can serve you.

ONLY 1 MOVING PART!

Here's the patented principle of the R & M Moyno Pump's amazing performance. A single-threaded helical rotor revolves within a double-threaded helical stator, providing pumping action like that of a piston moving through a cylinder of infinite length.



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In Canada: Robbins & Myers Co. of Canada, Ltd., Brantford, Ont.

HOISTS • CRANES • MOTORS • MACHINE DRIVES • FANS • FOUNDED 1878

Topics

MIDGET SEARCHLIGHTS — only as big as a walnut yet so powerful they will project a 1500-candlepower beam visible for 60 nautical miles—have recently been designed to aid rescues of aviators forced down at sea. The new lamp provides the most powerful beam ever obtained from such a tiny incandescent source. The unit has a molded plastic case and dome making it low in weight, a necessary factor inasmuch as the light is worn on the head to enable the man to have both hands free in rough weather. The six-watt lamps are wired to a small hand-cranked generator which is included in the raft equipment to supply power for a radio. The lamps can also be flashed as signal lamps by means of a "trigger switch."

PLASTIC BAYONETS are a new development of the Navy designed for drill and parade ground use, releasing thousands of steel bayonets for service on the fighting fronts. Molded from highly impact-resistant phenolic resin board, the bayonet blades with metal insert handles are being produced in a two-cavity compression mold.

NYLON TIRES, while still in an experimental stage with some problems remaining to be solved, meet the most severe form of the standard bruise test without breaking. This test involves pushing a blunt cone-shaped plunger down upon the tread of an inflated tire until it bursts. On a regular 9-inch tire the plunger had sunk six inches before bursting occurred, while on a similar nylon tire the plunger was forced inside the tread clear to the rim without breaking the tire; when withdrawn, the tire sprang back without injury.

RECENTLY REVEALED but "time tried," according to the company, is a successful hot-dip process of coating iron and steel parts with lead, conserving scarce zinc normally used for these coatings.

"SYNTHETIC LUMBER" is now being produced from gypsum rock for planks, wallboards, exterior boards and panels of many types.

A TREMENDOUS JOB of conversion to the manufacture of war materials has been faced due to our aid to the United Nations and our subsequent active entry into the war. This conversion program involved, for instance, redesign of thousands of parts to make possible their production by the economical, speedy, stamped metal processes that had been successfully used in producing civilian goods. Because it was felt that only by united effort, exchange of data, and pooling of experience could maximum results be obtained, the Pressed Metals Institute was organized by sixty representatives of sheet and strip metal fabricators, rolling mills and metal press manufacturers serving the industry. Objectives are to co-operate closely with appropriate government departments to speed up and increase mass production of war materials, and to engage in research to develop new and extended uses for metal stampings in the postwar period.

KOK-SAGHYZ, the Russian rubber-producing dandelion, is now in experimental stages in the United States. On government-owned land acres of the dandelion have been planted. The main objective is to secure the seed, although a small portion of the crop may be harvested and subjected to experimental processing for the extraction of rubber. Planting of seeds received direct from Russia has indicated a yield of 50 pounds of rubber per acre. Advantage of the dandelion over other rubber-producing plants is that it can be harvested and processed in a single year.

SPLIT-SECOND TIMING in processing plants such as those producing synthetic rubber, aviation gasoline, and toluene for explosives, is done by a "robot" control which opens and closes dozens of valves—another example of automaticity as a vital war aid.

REPLACING RUSSIA'S war-damaged power facilities in small war plants behind the lines with a large number of small diesel engine-driven electric generating units is a part of America's lend-lease program. Recently completed was a delivery of a $4,000,000 order for diesel engines which represent a new development in the supercharged two-cycle field.

PLASTICS PRODUCTION is growing more rapidly on the West Coast than in any other part of the country. It has increased more than 500 per cent during the past two years, with military aircraft producers being the largest consumers. More than 200 parts go into an average ship.

Cartoon, courtesy Adel Precision Products Corp.

Specify Standard Parts!

By Colin Carmichael

FREEDOM from regimentation in design as well as in other fields of endeavor is one of the prerogatives to which our nation's designers will cling, come what may. Because of it the designs of our war machines have not been "frozen" to the same extent as those of our totalitarian enemies and we have been able constantly to improve the qualitative superiority of our fighting equipment. With manpower and other resources of the opposing nations more evenly matched than some of us realize, speedy victory depends largely on maintaining and increasing our technical lead.

Zeal for improvement, however, must not be permitted to create a demand for complete design flexibility down to the smallest detail, lest we find ourselves left behind through diminished production and inefficient utilization

of resources. Freedom in design does not imply independent action without regard to the rights of others or to the national welfare, but carries with it an obligation to consider the mutual advantages of cooperative effort.

Recognition of these advantages already has brought substantial benefits to the design field in the form of standardized machine elements not only on a commercial scale but also on an industrial, national and even international scale. In the formulation or adoption of officially recognized standards the individual designer as a rule, can exercise only an indirect influence. However, in the encouragement of commercial standards through intelligent adoption, wherever possible, of generally available commercial machine parts he can make a real contribution toward conserving and exploiting to the full our resources in manpower and production. Such standard parts may be defined as components which, though not recognized by any industrial, national or international standardizing agency, are manufactured in such quantities and are so readily available commercially that in effect they possess most of the advantages of official standards.

Many machines, standard or special, are little more than ingenious assemblies of basic parts arranged in a particular manner. In one type of equipment, for example, where a conscious effort was made to utilize stock parts it was found possible to cover a wide range of sizes and styles using 90 per cent stock items, somewhat in the manner of an "erector set". Practical considerations of strength, inertia, thermal loads, etc., however, limit the application of this principle so far as most machines are concerned.

Advantages to the user and service man of employing such standard parts are obvious and the benefits enjoyed by the manufacturer are equally great. If one part can be made to serve where two were formerly required, the higher production rate enables better quality and greater uniformity to be obtained at the same or lower cost. The problem of stocks also is simplified, helping to reduce shipment times, idle inventories, clerical records and manpower.

Progressive companies recognize that good design of a line of machines requires standardization of elements and details so that the maximum variety of complete machines can be assembled from the fewest possible variety of parts. In order that no important advantages be overlooked, a systematic approach is essential. It is the practice of the General Electric Co. to consider the following chief steps in addition to other considerations:

"First, the features common to all parts of a given type are agreed upon, including styling, finishes, tolerances, materials, and processes, so that in all shop drawings and shop operations as few differences as possible will exist. For example, in making bearings for small turbines a single drawing is made with details common to all sizes filled in and blanks left for the major dimensions. When any new bearing is required, filling in the blanks gives a drawing that departs the least possible from bearings already made.

Uses Preferred Numbers for Sizes

"Second, in laying out a line of machines, the preferred number principle is used to select the fewest possible number of sizes to cover the desired range. Often alternate sizes only will be kept in stock and the intermediate sizes will be made when required. This plan allows new designs to be added without disturbing the general plan of standardization.

"Finally, the completed machines are designed so far as possible to have a basic frame or skeleton suitable for a wide range of applications, with designs such that special parts or finishes can be added to suit the different service requirements. In other words, each different field of service is provided for by adding or changing parts or protective features rather than by changing the basic designs."

A typical example of modern design in which full consideration was given to the standardization of parts is found in the recently developed line of Westinghouse gearmotors, *Fig.* 1. In the previous line there were listed 27 sizes of gearcases with 38 output speeds involving 27 different ratios for each horsepower rating and requiring 361 different sets of gears. A.G.M.A. standardization of output speeds alone effected a reduction in the number of gear sets to 246. Then by standardizing center distances it was possible to reduce the total number of gear sets to only 147. This was accomplished by redesigning the various types of units in such a way that the gearing for a certain center distance in a single reduction unit could be used also as the high-speed set, for example, in a double-reduction unit having the same center distance, *Fig.* 1.

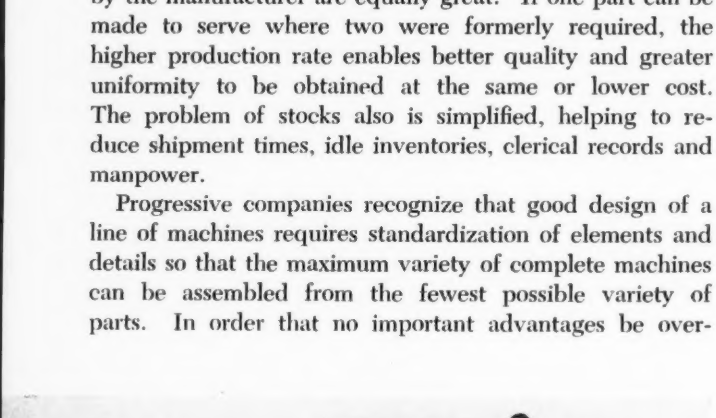


Fig. 1—Standardization on center distances for the gears enabled identical gear sets to be used in single-reduction (left) and double-reduction (right) units

In adopting standard components for incorporation in machines the designer has the choice of making up a "company standard" or of investigating commercial sources of supply of parts such as will be needed. The "company standard" parts may be made in the machinery manufacturer's own shops or may be bought to special order. A little reflection leads to the conclusion that this so-called company standard is no standard at all except in the few cases where the number of units needed is exceedingly large. Production rates cannot be as high as with commercial standard parts already being made and supplied to other users. Unnecessary duplication of wartime production facilities ties up machines which could well be released for other work.

Extensive utilization of standard commercial parts may not appeal strongly to the designer's creative instincts.

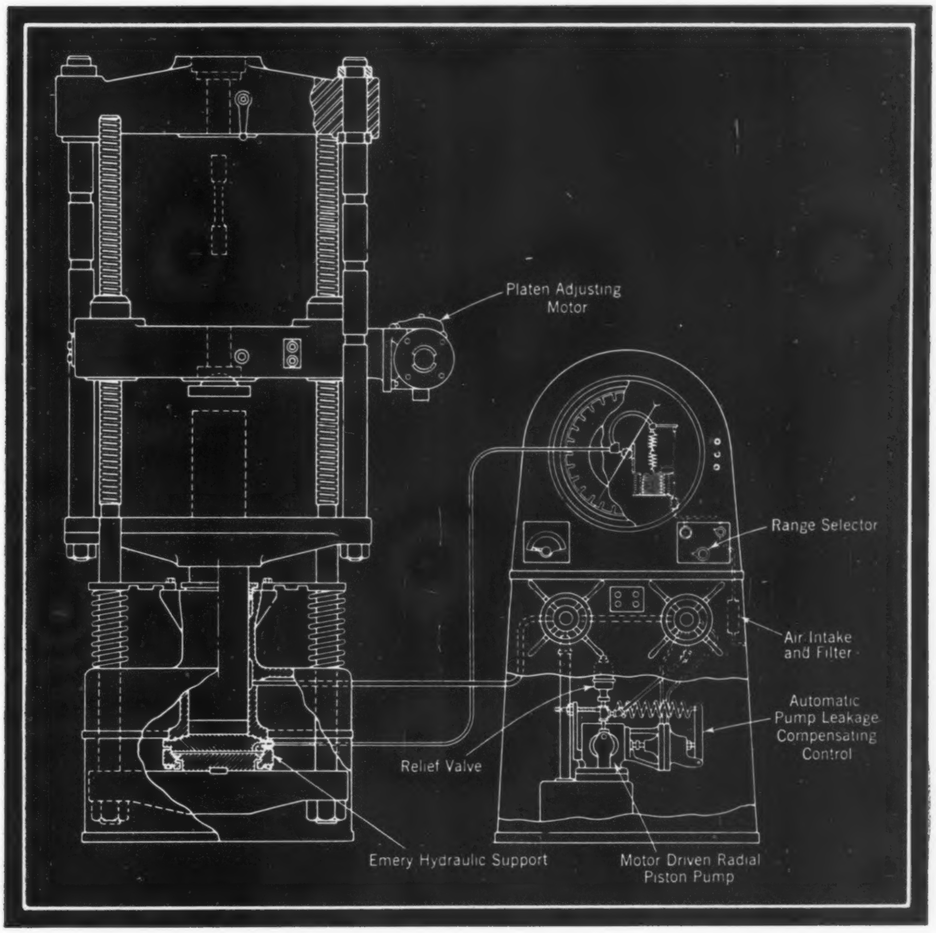


Fig. 2—In addition to standard commercial parts indicated on drawing, many smaller parts such as plastic control knobs are used in this testing machine

It may entail his modifying what he considers the ideal design in order to accommodate parts designed by somebody else, and if properly carried out involves careful investigation of the relative merits of competitive units. But it is sound engineering.

A notable instance of the application of these principles is found in the Southwark Tate-Emery line of universal testing machines, *Fig.* 2. The production of these machines is large enough so that any of the parts for the most used sizes could be considered standard in the sense that they are manufactured in quantities and used in the construction of testing machines by production assembly methods. In *Fig.* 2 there are indicated some of the standard components. Commercially obtainable parts include the radial piston pump for operating the hydraulic ram of the testing machine, the geared head platen-adjusting motor which replaces a formerly used gear arrangement, relief valve, air intake filter, air pressure reducing valve, three flexible shafts, a number of plastic control knobs, also the electric motors and their controls—magnetic switches and pushbutton controls. In connection with the electrical controls which are standardized by their makers the machinery manufacturer chose one of the recommended standard types.

Other parts which could not be obtained on the market were designed with a particular view to their suitability for use in a wide range of machines. Thus the automatic pump leakage compensating control unit can be used on all of the pump sizes regularly used for testing machines. The entire load indicator has been so designed that by changes in only a few parts it can be used with testing machines of any size and capacity, either directly connected to the hydraulic ram pressure or in conjunction with the special hydraulic support shown in *Fig.* 2.

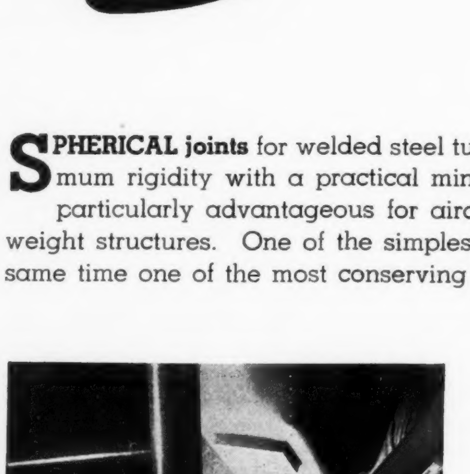
In one company where the policy of eliminating special parts wherever it is possible to use standard parts has been established, these parts include varied units such as gear reducers, grinding heads, high-speed spindles, drilling units, adjustable drill heads, differential drives, variable speed transmissions, pumps, and operating cylinders. These units have all the essential component parts built in and are ready to be put into service with little or no modification, while maintenance is simplified because all repair parts are standard with the supplier.

Through parts standardization sizeable amounts of material can be saved due to reduction in scrap as the result of eliminating the cut and try of straight job methods. Likewise only one set of tools is necessary for as many jobs as can use these parts, conserving tool materials as well as the time of the much-needed tool designers and tool makers.

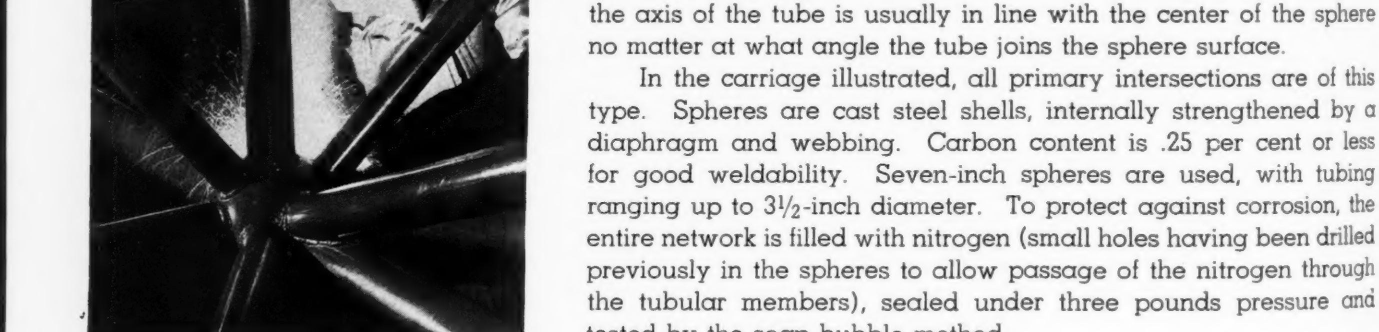
In all cases it is wise for the machine designer to consult with the parts manufacturer as to the present and probable future status of the parts under consideration so that a choice can be made that will insure a continuing supply at minimum cost. It is also well to ascertain how extensive are the service facilities and repair parts supply. Above all, inasmuch as each component becomes part of the machine for which the designer assumes responsibility, quality of standard parts selected must be at least equal to that of the machine as a whole.

Scanning the field for IDEAS

SPHERICAL joints for welded steel tubing requiring maximum rigidity with a practical minimum of weight are particularly advantageous for aircraft and other lightweight structures. One of the simplest designs and at the same time one of the most conserving of steel, this type of joint is utilized in the ship towing carriage, below, designed by Air Reduction Sales Co. The carriage has a 28-foot span yet weighs only 8 tons. Obviating cutting to fit the contour of adjacent members, most tube ends may be cut square since the axis of the tube is usually in line with the center of the sphere no matter at what angle the tube joins the sphere surface.



In the carriage illustrated, all primary intersections are of this type. Spheres are cast steel shells, internally strengthened by a diaphragm and webbing. Carbon content is .25 per cent or less for good weldability. Seven-inch spheres are used, with tubing ranging up to $3^{1}/_{2}$-inch diameter. To protect against corrosion, the entire network is filled with nitrogen (small holes having been drilled previously in the spheres to allow passage of the nitrogen through the tubular members), sealed under three pounds pressure and tested by the soap bubble method.



Maneuverability, speed and simplicity of control for fire-fighting equipment is highly important because a few seconds may mean success or failure in the rescue of plane personnel from airplane crash fires. For this reason controls for the boom nozzle, front nozzle and linear ground-sweep nozzle in the

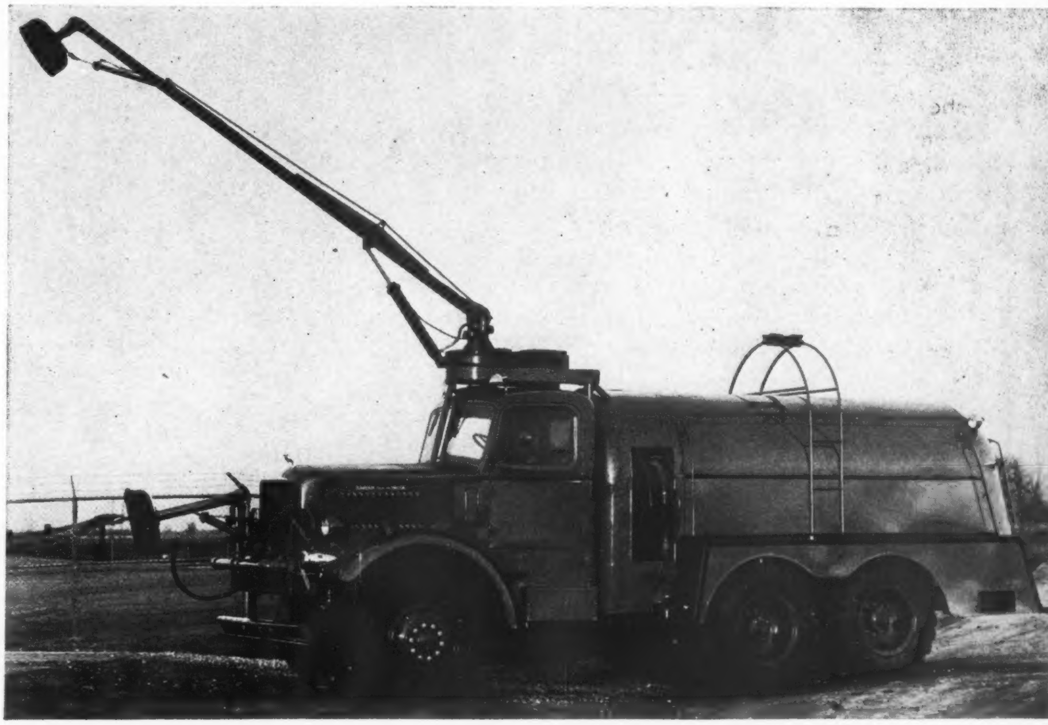
airport fire truck, below, all are mounted within easy reach of the driver.

With one movement of a control "joy stick" for any nozzle, a switch contactor starts a hydraulic motor and pump, another contactor operates a hydraulic-ram oil switching valve which causes the nozzle to travel in the desired direction. Boom and front-nozzle lifting devices as well as front-nozzle rotation are hydraulically actuated, while boom-nozzle rotation is effected by electric motor and gear drive. Each of the movements has extreme positions controlled by limit switches.

Electrical system of the truck, designed by the Cardox Corp., is all-important and it is therefore essential that the batteries be fully charged at all times. Energy is furnished by two heavy-duty diesel type batteries in an electrical system separate from the truck engine battery, connected in series to give a potential supply of twelve volts. A selenium rectifier, constantly connected to the batteries, is utilized for charging when the truck is at its station. This rectifier is controlled by a temperature-compensated relay and a high-low charge control relay.

Because the energy supply is disconnected when the truck leaves the station, motive power for the relay-controlled rectifier has been eliminated. To place the relays in proper contact for a high-rate charge immediately upon return of the truck to its station, a normally open, spring-loaded charge-reset relay is employed. This is connected direct to the hydraulic pump motor so that any operation of the hydraulic system will place the rectifier in high-rate charging position.

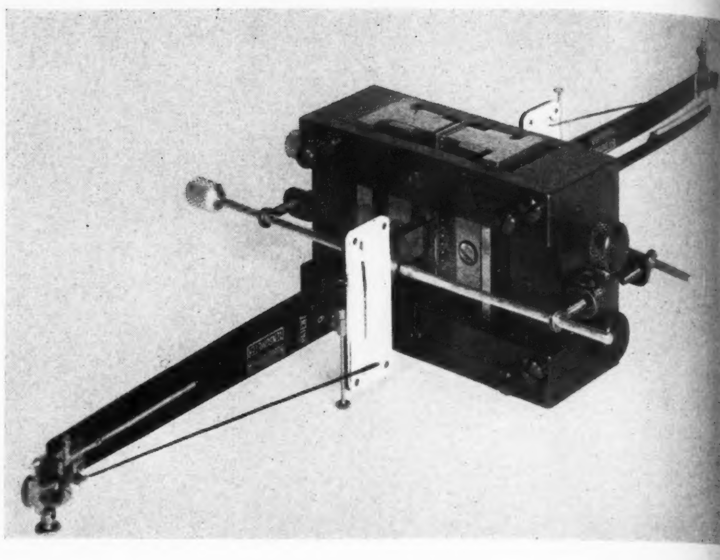
Hose reel on side is so arranged that when the operator lifts the gate to take the play pipe, a valve is operated which admits carbon dioxide—the fire-extinguishing medium—into the hose reel. Discharge is controlled by a valve mounted in the play pipe itself.



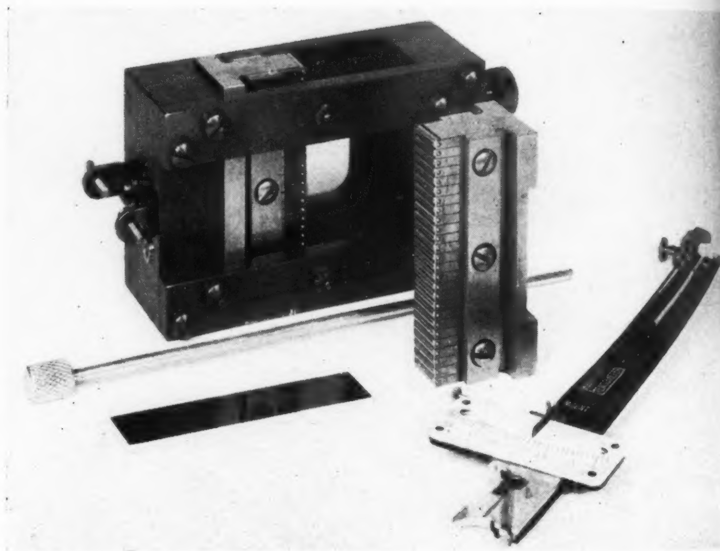
Carbon dioxide is maintained in liquid state by mechanical refrigeration at the uniform low temperature of 0 degrees Fahr. and the correspondingly low pressure of 300 pounds per square inch. Optimum rate of discharge for boom or front nozzle is more than a ton of carbon dioxide per minute.

Rust preventive for machinery and parts, eliminating the necessity of slushing and degreasing operations, protects hard-to-reach interiors as well as other assemblies in war equipment. Granular silica gel which is capable of absorbing half its weight in moisture is packaged with the parts requiring protection as shown at left. Humidity indicator cards developed by the Permutit Co. give visual indication at all times of the moisture-

absorbing properties of the chemical. Impregnated with cobalt chloride, the granules turn from blue to pink when moisture has increased above the safe 20 per cent value. Color card on the indicator identifies at a glance whether the machine parts are safe from corrosion.



Compression-yield testing of single sheets instead of prearranged packs allows the use of smaller loads and greatly simplifies testing technique by eliminating composite factors which are always present in pack testing. At right is shown a jig, developed at the Aluminum Co. of America research laboratories, which comprises two blocks with small rollers mounted on the face of each. Held in place by small, brass leaf springs, the rollers are chamfered on the ends in much the same way as some types of needle bearings. When pressure is exerted on the specimen being tested the rollers are free to move up or down because the chamfered ends push the springs out of position, permitting slight controlled movement. Tensometer on specimen measures the deformation, providing accuracy within one per cent, about the same as is available in tensile tests. Because a single sheet specimen is under constant support from the many rollers which are maintained with their axes constantly in line, a set of uniform conditions is provided that was not always available previously.



Flexibility of molded cellulose acetate butyrate allows for simplification of hinge design on cover for megger instrument as shown below. Bezel, molded of this material, has two lugs on opposite sides which may be snapped into sockets in the cover as shown in the illustration. When the bezel is attached to the flat surface of the molded case, the lugs are held in position. Thus the assembly, designed by Chicago Molded Products Corp., makes a simple and practical hinge without the use of metal.

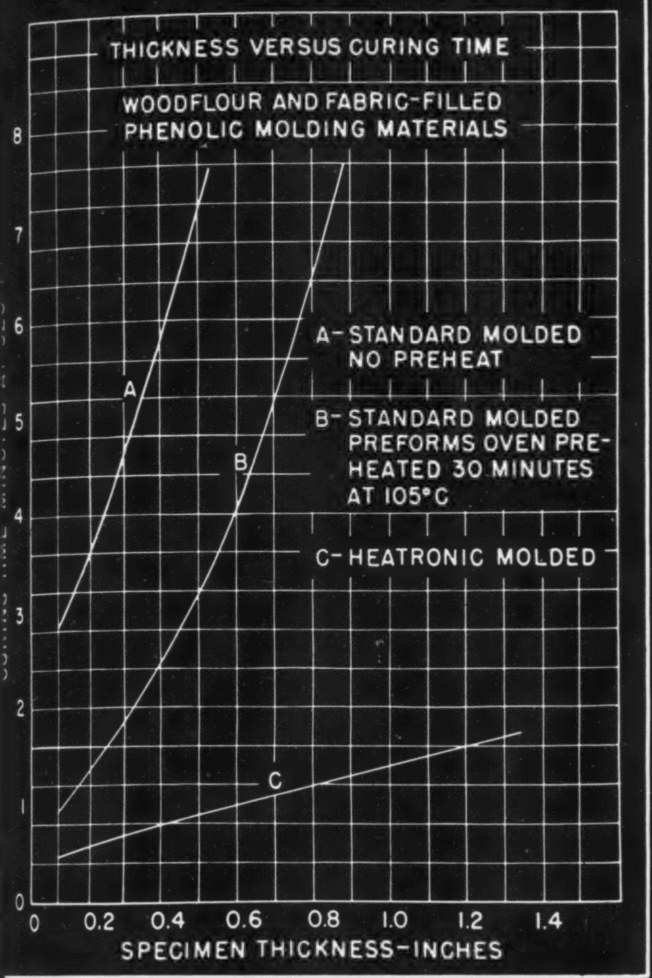


Fig. 1—Shows optimum time required for molding parts of different thicknesses

High-Frequency Heating Speeds Plastic Molding*

By V. E. Meharg
Bakelite Corp.

APPLICABLE to many phases of the forming or molding of plastics, the Heatronic or high-frequency heating process covers broadly a method of heating a plastic to a temperature at which it possesses sufficient fluidity for the desired flow. Because plastics are well adapted to making a large number of duplicate parts, the matter of speed in molding plastics has received the most intense attention. Variables of temperature and pressure have been used to the utmost in shortening cycles as much as possible. Today jobs are gained or lost by time cycle differences of only a few seconds. Inasmuch as plastics are subject to heating and cooling cycles in the molding operation the rate of heat transfer has been the most severely limiting factor. This is particularly true with thick sections, also with thermosetting materials because the complete operation of heating and hardening had to take place in the molding die. The high frequency method strikes at this problem by generating the heat within the material, and if desired, outside the die. It therefore removes the limitation which depends upon rate of heat transfer.

In its ultimate form the method consists of making the material the dielectric between two plates in a high-frequency field. In other words, the plates and the mate-

(*Continued on Page* 200)

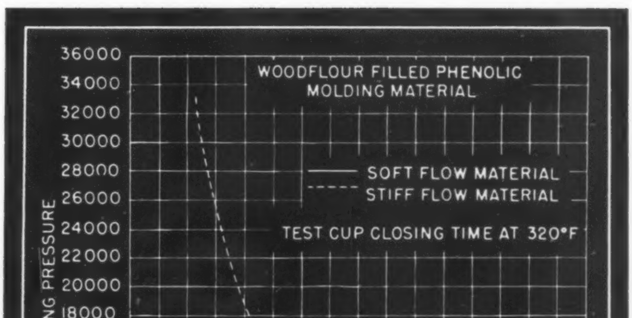


Fig. 2—Pressure required for molding as a function of closing time, is an indication of plasticity

*Abstract of paper presented at the recent annual meeting of the S.P.I. in Chicago.

Stress Concent on

By R. E. Peterson
Manager, Mechanics Dep't.
Westinghouse Research Laboratories

IN THE introduction to a recent publication on stress analysis appears the following statement: "This bulletin is not concerned with such questions as why stresses should be computed, or what should be done with the stresses after the calculations are made." The latter question, which usually is of no interest to the elastician, is one with which a designer must live and is the subject of this article. Specifically, the discussion deals with the relation between stress concentration factors obtained photoelastically or otherwise, and the strength of the corresponding machine parts. As will be evident, there is a great deal to be learned in this field, but it is worthwhile pointing out what is known and how this knowledge can be used in design.

To simplify discussion, it may be said that the effect of a localized stress caused by a notch, fillet, hole or similar "stress raiser" depends mainly on the *type of loading* and the *material*. The article therefore will be confined to the following cases, although it is recognized that there are other variables such as effect of temperature, time, etc.,

Fig. 1—Top—Successful design of turbogenerator rotor results from proper application of stress concentration factors such as photoelastic fringe patterns can furnish

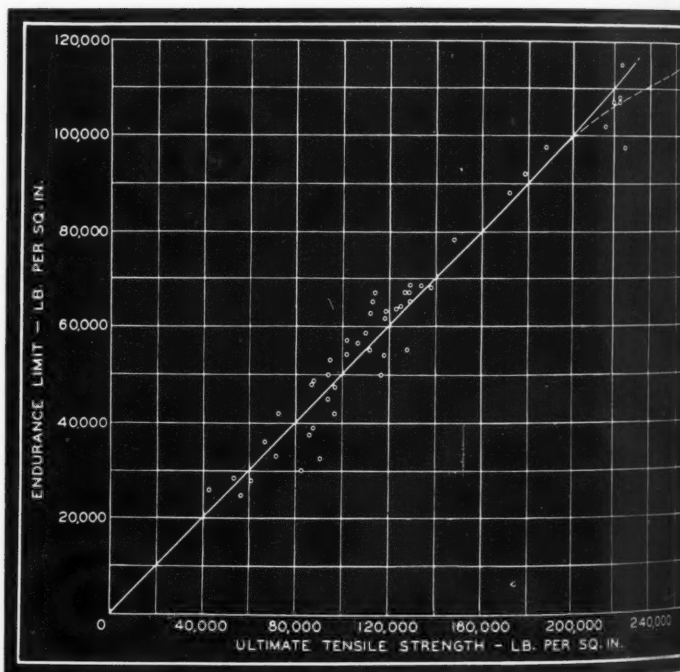


Fig. 2—Right—Relation between endurance limit and ultimate tensile strength is typical of results obtained in usual fatigue tests with no stress concentration (H. F. Moore)

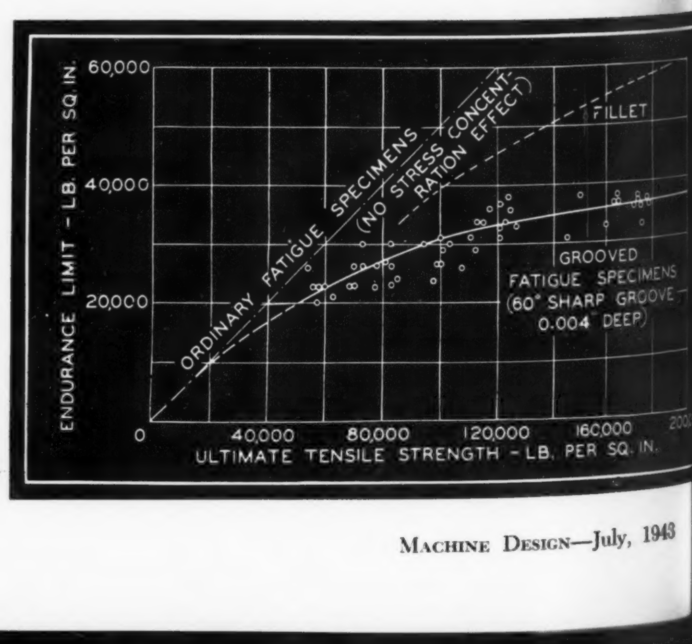


Fig. 3—Right Below—Curve shows that higher strength materials generally are more sensitive to stress concentration (Mailaender)

n Factors in Practical Design

and various combinations:

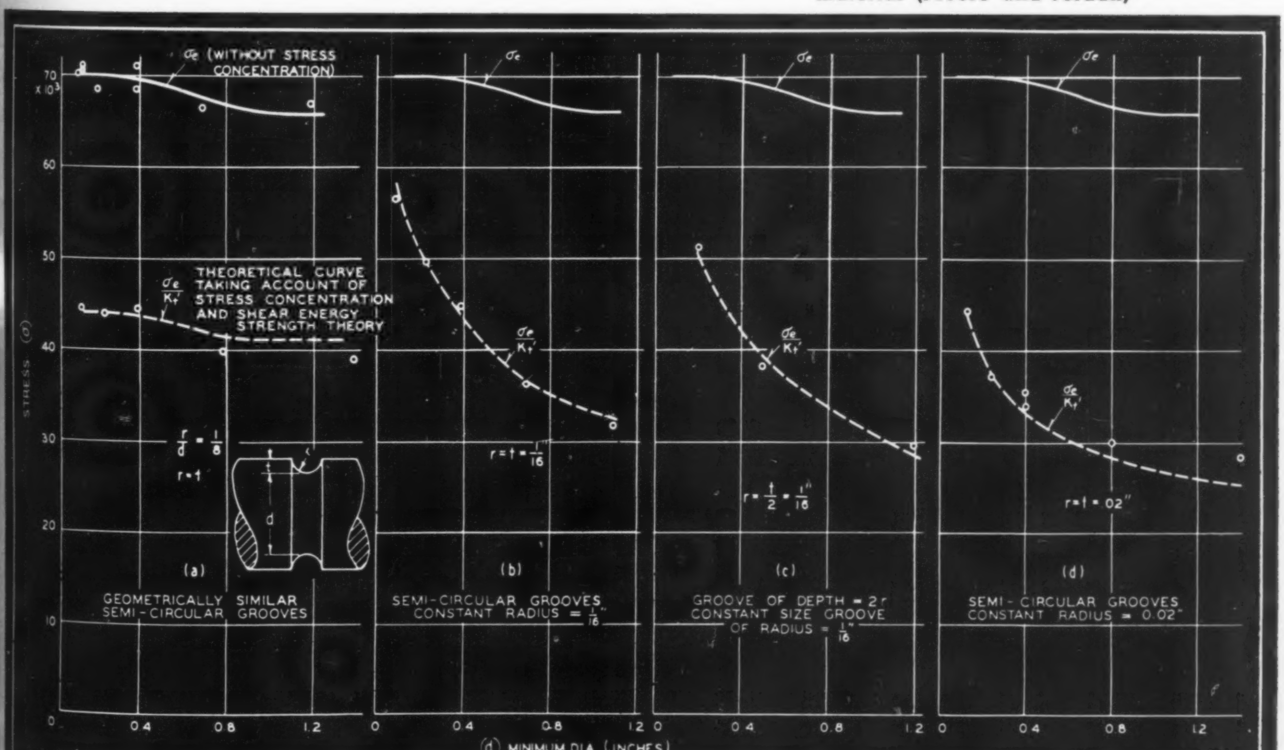
Brittle material
 a. Static loading
 b. Alternating loading
Ductile material
 a. Static loading
 b. Alternating loading.

Brittle materials, such as plain cast iron, are sometimes defined as those having an elongation of less than 5 per cent. Since such materials are generally not used to carry critical loads, it is unnecessary to include extensive discussion of this classification. Failure in brittle materials occurs in accordance with maximum normal stress. The strength reductions which are obtained with brittle materials *statically loaded* are usually not quite as great as indicated by application of stress-concentration factors. In the case of *alternating stressing*, brittle materials often show a surprisingly small reduction of strength for small grooves and notches. Moore has explained this result in the case of cast iron by pointing out that such material is full of stress raisers to begin with, in the form of the holes occupied by the graphite flakes. It is, however, customary in design to apply the full stress-concentration factors as determined photoelastically or otherwise for all cases involving brittle materials and it would not be wise to depart from this practice without detailed knowledge of the particular application in question.

Static Strength Not Reduced by Stress Raiser

Ductile materials, of course, comprise the major portion of engineering applications. For *static loading*, as typified by the slow tension test, a ductile material yields in the regions of highly localized stress so that the final rupture strength is not reduced by the presence of the stress raiser. There are other considerations, such as the importance of elongation or energy absorption in the case of failure, but as far as strength is concerned it is customary design practice not to apply stress-concentration factors in cases of ductile material statically stressed. This brings us to the most important application of stress-concentration factors, namely, the case of ductile materials wherein

Fig. 4—Below—Analysis of fatigue tests on SAE 2345. Dotted curves are theoretical, plotted points are test data, showing excellent correlation for this high-strength material (Moore and Jordan)



the loading, or at least a considerable portion of the loading, is alternating in character, so that fatigue of the material is the governing consideration.

Ductile steels, for example, vary all the way from soft structural steels up to some of the heat-treated alloy spring steels. It is well known that the effect of stress concentration is more severe in heat-treated steels than in normalized or annealed steels. This is what is often meant by the "tenderness" or "sensitivity" of certain high-strength materials, although these qualitative terms have also been used in a more general, and sometimes vague, sense. The effect to which reference is made is illustrated by *Figs.* 2 and 3.

PRESENTING an encouraging picture of the practical uses to which theoretical stress concentration factors can be put in actual design, this article is based on a paper given by the author at the seventeenth Photoelasticity Conference and Stress Symposium in Detroit. It supplements previous articles on stress analysis which have appeared in MACHINE DESIGN, of which the most recent are listed among the references on Page 108

As is generally appreciated, *Fig.* 2 shows that the endurance limit of conventional test specimens (where the contour varies so gradually that no concentration of stress occurs at the critical section) increases roughly in proportion to the ultimate strength. The line representing this relationship is repeated for comparison on *Fig.* 3, which shows that stress concentration has a greater effect for the higher strength materials. Curves of the kind shown in *Fig.* 3 differ widely with form of notch or stress raiser and also with size, as will be mentioned later. The important point to remember is that machine parts are practically never shaped like conventional test specimens, but contain stress raisers in the form of threads, fillets, grooves, keyways, splines, oil holes, etc., so that the fatigue strength of actual machine parts is more apt to be governed by a relationship like that of the two lower curves of *Fig.* 3, than that of the straight line of *Fig.* 2. This means that the percentage gain in the fatigue strength of an actual machine part attained by going to a high-strength steel is apt to be less than is often assumed.

Results Presented for Different Materials

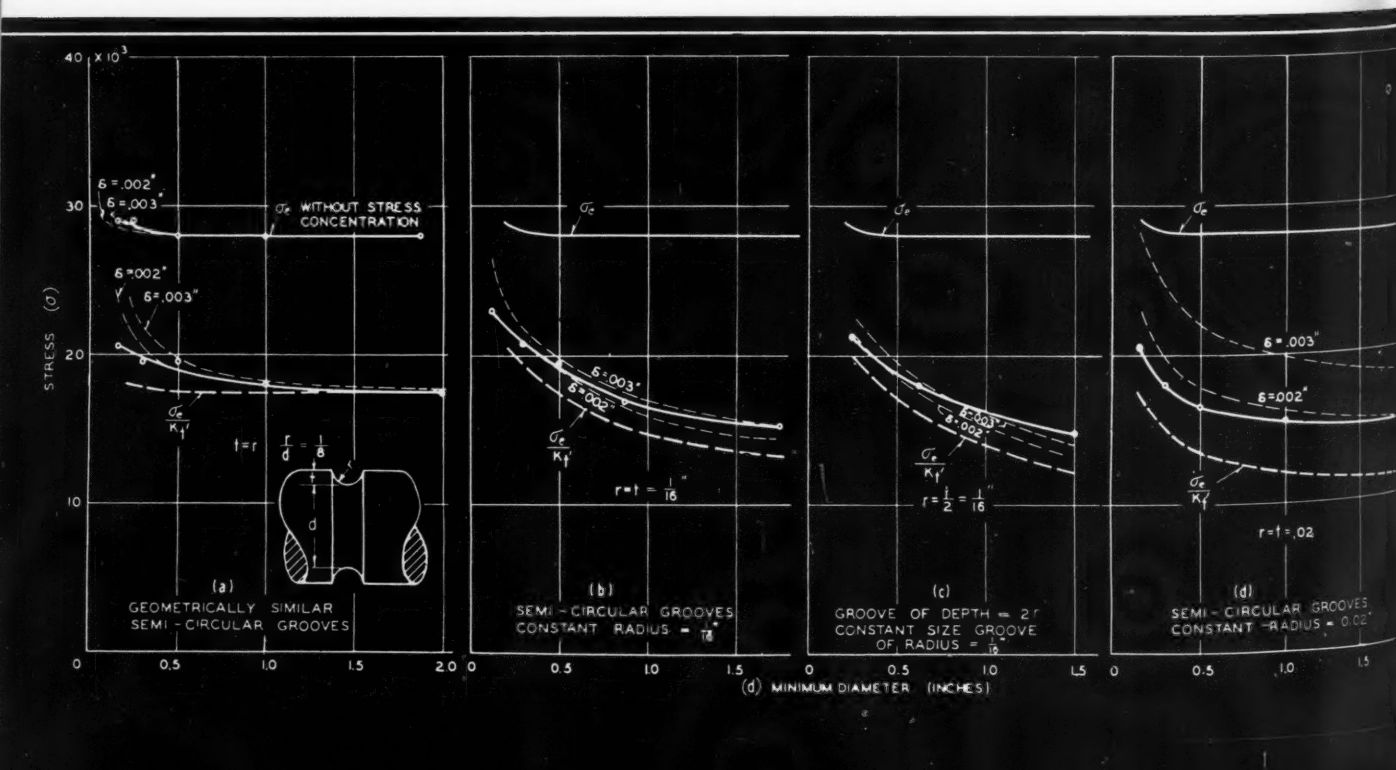
To illustrate in more detail the type of results obtained with different materials, the following analysis of data (5*a*)* for a quenched and tempered nickel steel (SAE 2345), *Fig.* 4, and for a low-carbon steel (SAE 1020), *Fig.* 5, is presented.

Referring to *Fig.* 4, the upper series of curves is drawn to fit endurance limit values for specimens of various diameters and without stress concentration. Note the following point carefully—each lower curve is in a sense a theoretical curve, since it was obtained by dividing values on the upper curve by theoretical factors K_t' (*Fig.* 6) which take account of stress concentration and the shear energy strength theory. The agreement of the lower curve with the plotted experimental points is remarkable, considering the range of groove size and specimen diameter.

Data on SAE 2345 and similar data (Ref. 5*b*) for high-strength heat-treated steels show that at least for one class of material there apparently is at hand the means of making a good prediction of the fatigue strength of complicated shapes and parts. The fact that such good agreement between theory and experiment is obtained for this class of material, as in *Fig.* 4, is encouraging since it holds out a hope that such analysis can be extended to cover a larger portion of the materials field. It would, indeed, be a bleak outlook if the only way to enlarge the

*References in parentheses are listed at end of article.

Fig. 5—Analysis of fatigue tests on SAE 1020 shows correlation between theoretical and experimental endurance limits at small distances below surface of material (Moore and Jordan)



field of knowledge was by endless testing, without the benefit of theoretical guidance. This is not to argue against testing of materials or life testing of assembled apparatus, of which too much cannot be done at present. It is a way of saying, however, that the research work of the elastician, the photoelastician and the strain gage expert will be increasingly used by the design engineer as technical knowledge advances.

Referring to *Fig.* 5 for the low-carbon steel (SAE 1020), it is seen at once that the reduction in fatigue strength is generally not as great as that indicated by the theoretical factors. It should, however, not be overlooked that for the geometrically similar specimens it appears that as specimen size increases a limiting condition is approached where agreement is reached. As to the reason for this behavior there is no complete answer, although considerable work has been done. Considering a series of geo-

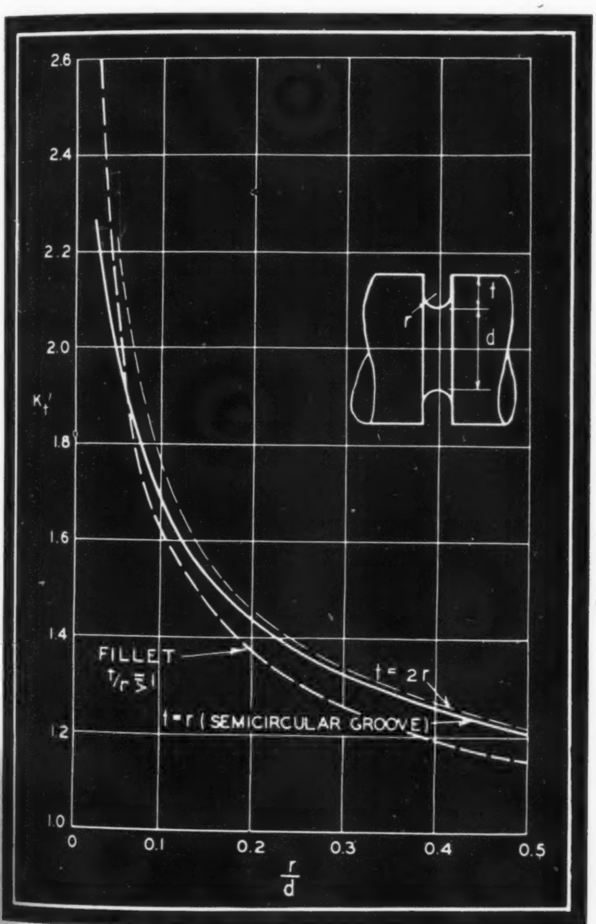


Fig. 6—Theoretical stress concentration factors for semicircular groove are based on shear energy theory

metrically similar specimens of varying size (*Fig.* 5a), it is useful to list the possible departures from similarity. These are: (1) *Grain size*, which is constant, but not geometrically similar in different size pieces; (2) *Stress gradient* of peak, that is, drop in stress per unit distance below surface; (3) *Machining* effects, including finish and cold working.

A great deal could be written concerning these variables but only one analysis will be mentioned here, that based on stress gradient. If it is assumed that failure occurs

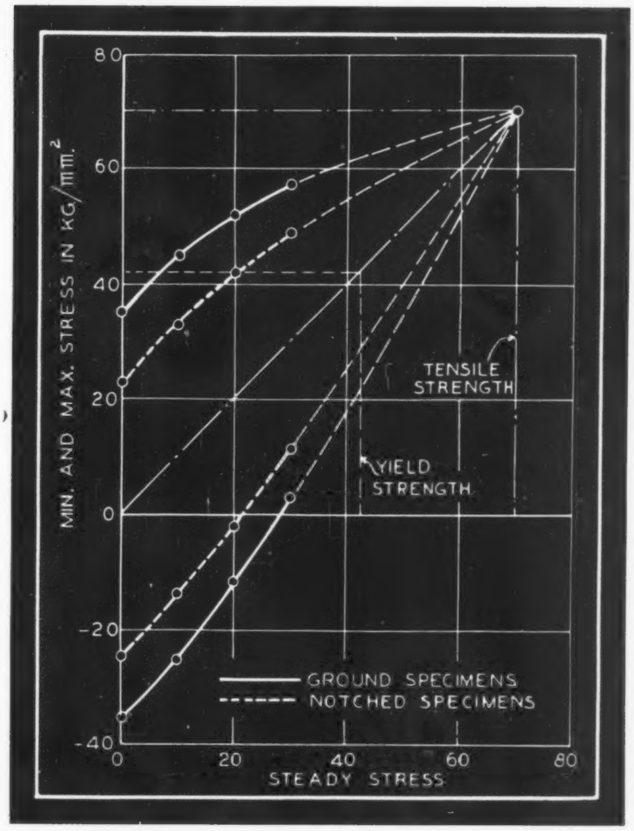


Fig. 7—Notch effect in fatigue with initial tension, for .70 per cent carbon steel. Curves show that for combined stress the stress concentration factors should be applied to the alternating component only

when the endurance limit is reached at a constant distance, δ, beneath the surface, the dotted curves shown in *Fig.* 5a are obtained. It is seen that in a general way these curves have the same trend as the experimental ones. The value of δ which appears to fit best in this case is .002-inch. This happens to be approximately the grain size for this particular material, although in other tests δ was found to be several times the grain size. Another possibility is that δ is due to the cold work of the machining process. As stated previously, there is as yet no complete answer to this phase of the problem.

Alternating Component Is Primarily Affected

Only a relatively small amount of research has been done for cases of combined stress involving stress concentration, although such cases are important in engineering practice. Referring to *Fig.* 7, it is seen that a range of stress may be considered to be made up of static and alternating components and that stress concentration seems to affect primarily the alternating component. In fact, it is customary in design to apply theoretical factors to the alternating component of stress and not to the static component.

The application of theoretical factors may be summarized for the design engineer as follows:

For brittle materials failure occurs in accordance with maximum normal stress, and full theoretical stress-concentration factors, obtained photoelastically or otherwise, should be applied to all cases of static, alternating or com-

bined stress.

For ductile materials, as far as static strength is concerned, it is not customary to apply stress concentration factors, although as mentioned there are sometimes other considerations to keep in mind.

In the case of alternating stress, the behavior is not the same for different kinds of ductile materials. For the heat-treated high-strength steels, the full theoretical factors of the kind shown in *Fig.* 6 (taking account of stress concentration and shear energy theory) should be applied and, as previously stated, a good estimate may be expected. For untreated steels, the same theoretical factors may be used if no other data are available and in this case it is of interest to note that whatever error is made is on

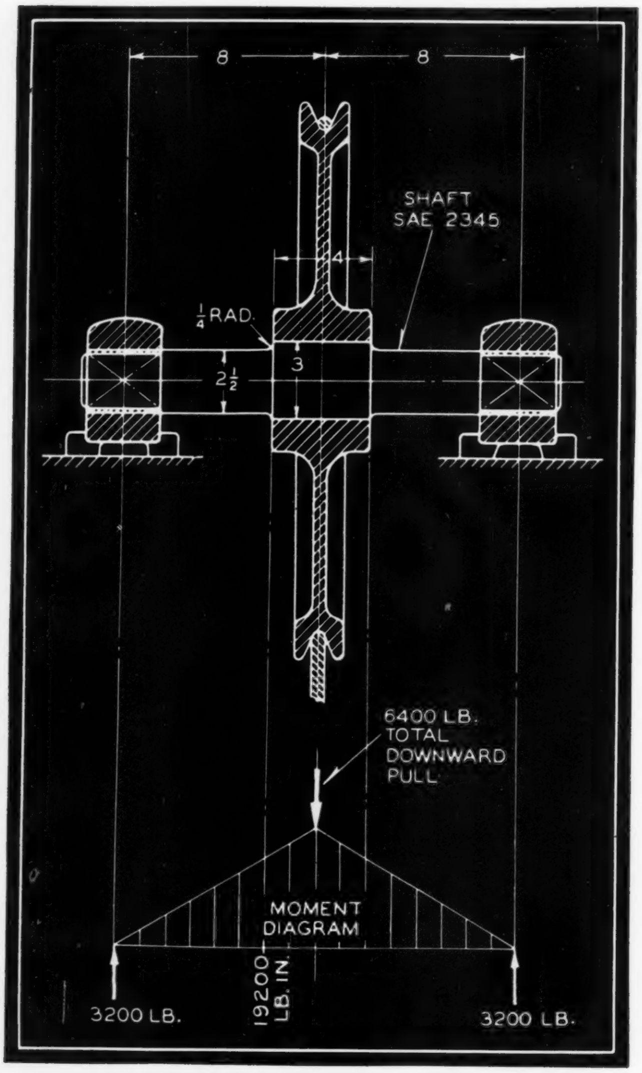


Fig. 7—Elevator sheave shaft provides typical example of how to apply stress concentration factors

the safe side. A rough correlation method is given on Page 181 of the Timoshenko Anniversary Volume published by MacMillan in 1938; this method may be useful in some cases.

For combined static and alternating stress in ductile materials, it is customary to apply theoretical factors to the alternating component in accordance with the foregoing, but not to apply them to the steady component.

In conclusion, it is perhaps well to emphasize that most machine parts subjected to severe stress are made of high-strength heat-treated steels and that for this most important case we are fortunate in having all the information necessary for accurate solutions to design problems.

Example Illustrates Application

To show how the type of information presented in this article can be applied in practical design the following example is given. The problem is to find the factor of safety of the elevator sheave shaft shown in the accompanying illustration, *Fig.* 7. The shaft is subject to an alternating stress in bending, the maximum occurring at the change in diameter at the edge of the sheave hub.

Solution: First find the nominal bending stress in the shaft at the edge of the sheave hub:

$$\sigma_n = \frac{Mc}{I} = \frac{32M}{\pi d^3} = \frac{(32)(3200)(6)}{(\pi)(2.5)^3} = 12{,}500 \text{ pounds per square inch}$$

Now multiply by the proper stress concentration factor, K_t'. From *Fig.* 6 it is found that for $r/d = .25/2.5 = .1$, the fillet factor is 1.6. Since this is a heat-treated alloy steel the full factor applies. The fillet stress is therefore:

$$\sigma_{max} = (12{,}500)(1.6) = 20{,}000 \text{ pounds per square inch}$$

For SAE 2345 the endurance limit (from *Fig.* 4 for shaft diameter in excess of 1.2 inches) is 66,000 pounds per square inch. The factor of safety is therefore:

$$n = \frac{66{,}000}{20{,}000} = 3.3$$

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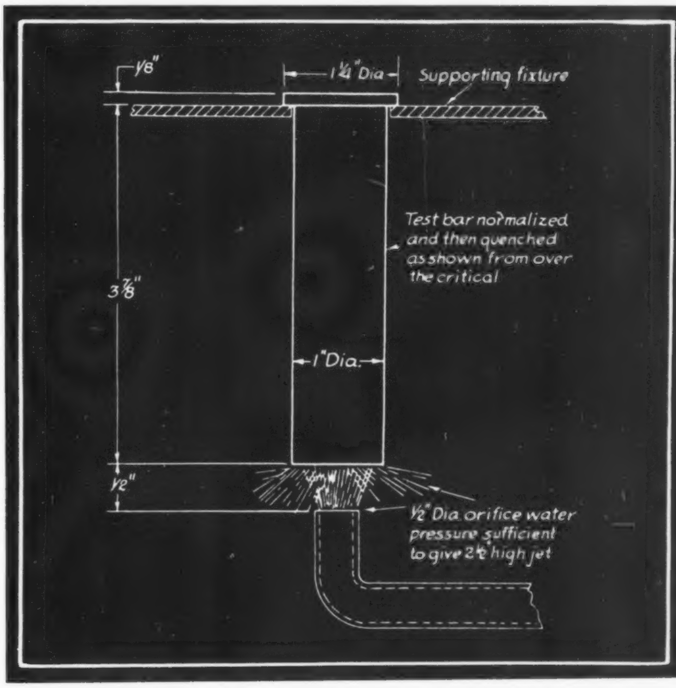
Fig. 94—Standard Jominy hardenability test. Hardness traverses are made on .015-inch deep flats ground lengthwise on diametrically opposite sides of bar



Wartime Metallurgy Conserves Strategic Materials

Part XII—Effects of Alloying Additions

By R. E. Orton and W. F. Carter
Acme Steel Co., Chicago

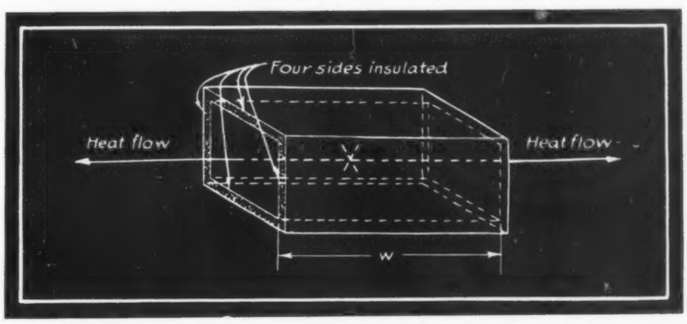


Fig. 95—Unidirectional heat flow is approached at the center of a large plate

ALL FEATURES of ferrous metallurgy of significance to the machine designer have been reviewed in this series. Effect of alloys introduces no new variable or characteristic of import, but modifies simply those metallurgical features already discussed. How these modifications occur, and their quantitative effect, will be the theme of this and the next article. Such being the case, previous articles will be reviewed briefly.

The prime purpose for the selection of one material in preference to another is to secure the best possible insurance against failure consistent with the economics of the application. To evaluate this insurance the physical tests described in the introductory article were designed. While not by any means conclusive, they have gone a long way toward facilitating a prediction of the performance of a material. They are the bridge, so to speak, between the material and the design.

Bain (1)* and many others have demonstrated that these physical qualities are a function of the carbon content and the metallographic structure, rather than of any alloying content. Accordingly, the structure of metals was carefully examined in succeeding articles.

Structures of Metals

Regular and orderly arrangement of atoms characteristic of crystals was discussed, as well as the directional properties inherent in such an arrangement. Cohesive failure by rupture of atomic bonds represents the ultimate in strength and brittleness. The more usual failure is by slip on atomic planes, and metals are strengthened by keying these planes. Emphasis was then placed on the dimensional limitations of the crystal form, and the

*References in parentheses are listed at end of article.

sporadic arrangement of crystals which result in a "statistical effect", physical properties being due to an averaging of many crystals.

A discussion of the structure obtained by "slow" cooling from above the critical temperature range, eutectoid pearlite, excess ferrite and excess carbide was then presented. Dispersed carbide structure obtained by cooling rapidly to some temperature below that of the "pearlite gate" of the S-curve was also dealt with. It was explained that if the cooling is continued to below some 300 degrees, martensite is produced, the hardest form of steel because it has the maximum of dispersion of the carbide particles. Tempering permits of some agglomeration of these particles with attendant softening to a more ductile, but weaker, structure. It also relieves the quenching stresses.

Hardenability, the "ease of obtaining a martensitic structure" was also covered. The full significance of this characteristic is reflected in the alteration of the isothermal curves, and the practical measure of it is made by the Jominy end-quench. Later articles in the series brought out the significance and the control of grain size and gave a resume of heating furnaces, attendant equipment and quenching and tempering operations. Production and classes of iron and steel were detailed. An account was given of the impurities unavoidably present in the commercial product, and of the small quantities of assisting elements generally added. This was followed by an exposition of the case-hardening processes, in the May and June issues.

Effects of Alloys Are Similar

Probably more has been written on the influence of alloying elements than on the remainder of metallography and heat treatment of steel. At first glance there appears to be no order to it all. Yet so similar is the effect of nearly all the alloys that the great bulk of the story can be told for all as a group, changing for each only the percentage. The choice of alloys may be based on practical considerations, economy, availability, machinability, ease of heat treating, etc. A realization of the practicality

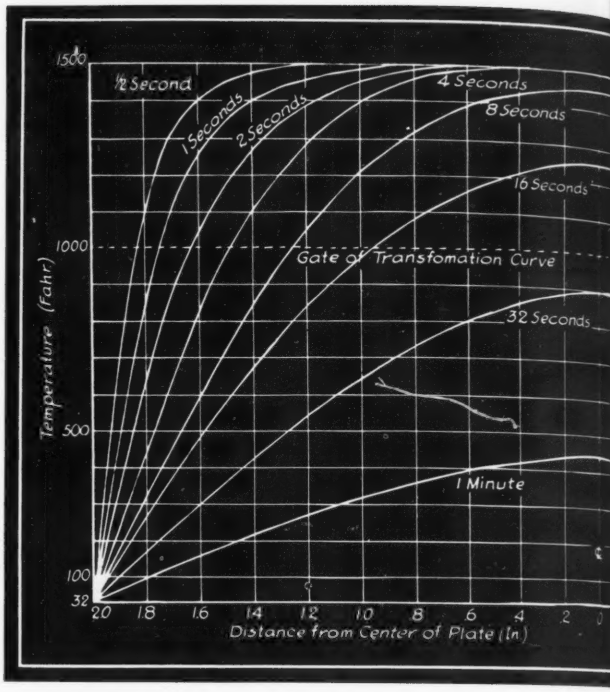


Fig. 97—Temperature gradients across a four-inch thick plate for an ideal quench, from 1500 degrees

of this approach is necessary in the war emergency in order that the country's resources may be utilized to the full with a minimum of wastage.

A broad division in the major alloying elements may be made according to their ferrite solution tendencies. The "ferrite formers"—nickel, phosphorus, copper and silicon—do not combine at all with the carbon, being present entirely in solution in the ferrite and strengthening thereby the matrix and moderately increasing hardenability. All other elements may be present as carbides but may also dissolve in the ferrite, the carbide-forming or ferrite-dissolving tendency varying widely.

This gives the second group—molybdenum, tungsten and vanadium—with strong tendency to form carbides, little solution in the ferrite, sluggish in dissolving in the austenite and in softening on tempering. They may strongly increase hardenability, but only if well dissolved in the austenite and homogeneously distributed. Manganese and chromium constitute the third group, those whose tendency is about balanced. The distribution of these elements between ferrite and carbide depends upon the relative amount of carbon and alloying element present, and the degree of solution and homogenization in the austenite. They dissolve readily in the austenite, serving practically thereby as strong increasers of hardenability.

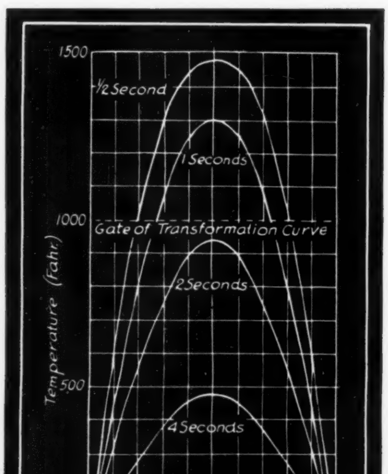


Fig. 96 — Temperature gradients across a one-inch thick plate for an ideal quench, from 1500 degrees

Steels Classified According to Use

Alloy steels may be divided according to the metallographic structure of final use. One group is used in what might be termed the "normal pearlitic" form "as rolled", annealed or, in larger section, normalized, having hardnesses ranging up to about 200 brinell. In the other group are the heat-treated steels, in general fully hardened to martensite and drawn back, but including also those treated by austempering and, in large sections,

those treated to obtain an emulsified pearlitic structure.

Steels in the annealed or as-rolled condition will not have the carbides favorably distributed. It is here that the effect of ferrite strengthening alloys is most pronounced in improving the physicals (2 and 3). This is the field of the low-alloy, high-tensile constructional alloy steels, of which there are some fifteen or more marketed under various tradenames. All of the ferrite formers are employed, as well as the mild carbide formers. In addition a small amount of moly and vanadium appears in some of the analyses, contributing undoubtedly to a finer pearlite distribution with its attendant improvement in physicals.

The strengthening effect of alloys in such a structure is mild compared with that obtained by heat treating. However, the requirements for such applications as bridges, buildings, railway cars, and so forth, are low cost, high ductility, good formability and weldability. To meet these requirements the ferrite strengtheners serve well. Whereas a yield of 25,000 to 35,000 pounds per square inch is common in structural carbon steel, these alloy steels will have yields ranging from 50,000 up to as high as 75,000 with little loss of plasticity or other qualities, permitting an increase in working stress of as much as 100 per cent.

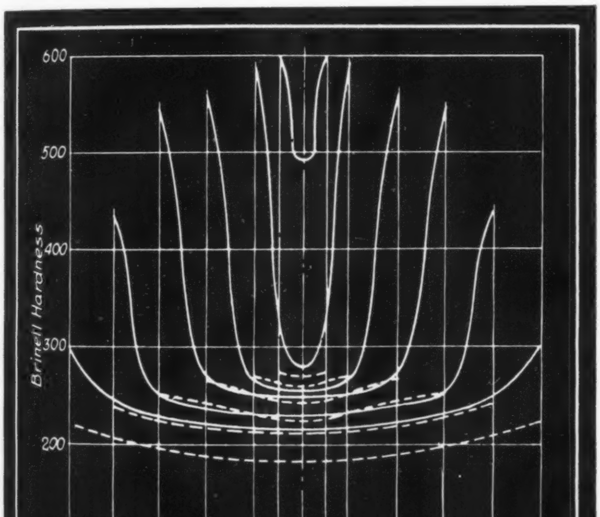


Fig. 98 — Above — Hardness traverses across various diameters of .4 per cent carbon steel rounds. Solid lines show water quench, dotted lines oil quench

In using these high-strength structural steels it should be remembered that the elastic modulus remains unaltered. This means that deflections may be higher than with a conventional design. Columns and other members where stiffness is important will not be able to avail themselves of all this extra strength. Machine frames and other structures where the design is based on deflection rather than stress will be benefited little by these steels. Where the design can be rearranged, however, to increase inertia of the section, there is much to be gained.

Since this series is concerned with the steels hardened by fairly rapid cooling rates, the remainder of this article will be concerned with this group. An analysis of the attaining of a metallographic structure from the discussion of hardenability in Part V may be made in terms of four variables:

1. Capacity of the steel for hardness
2. Hardenability
3. Cooling rate
4. Temperature and time of tempering.

The capacity of a steel analysis for hardness is determined by the amount of carbide particles that may be formed to key the atomic slip planes. This is a function of the *carbon content only*, the addition of alloys in the amounts usual in constructional steels making little contribution. This capacity, as measured by hardness readings on fully hardened carbon steels, is given in the curve of *Fig.* 32*.

Hardenability is the ease with which a molecular dispersion of the carbide particles may be secured. It is effectively measured by the Jominy test, *Fig.* 94, in terms of the cooling rate, *Fig.* 45†, required to suppress the formation of pearlite. The actual cooling rate existing at any point in a part is a function of the quenching medium and procedure, of the shape and mass and of the location of the point in the part. Tempering to the desired dispersion is of course only possible provided a finer dispersion has already been obtained in the quench.

It is seen that, of the four variables that enter into the determination of the metallographic structure of a processed part, only the second is appreciably affected by the introduction of an alloying element. It would also seem that, with the single curious exception of cobalt, every

*Machine Design, Nov., 1942, P. 84.
†Machine Design, Dec., 1942, P. 95.

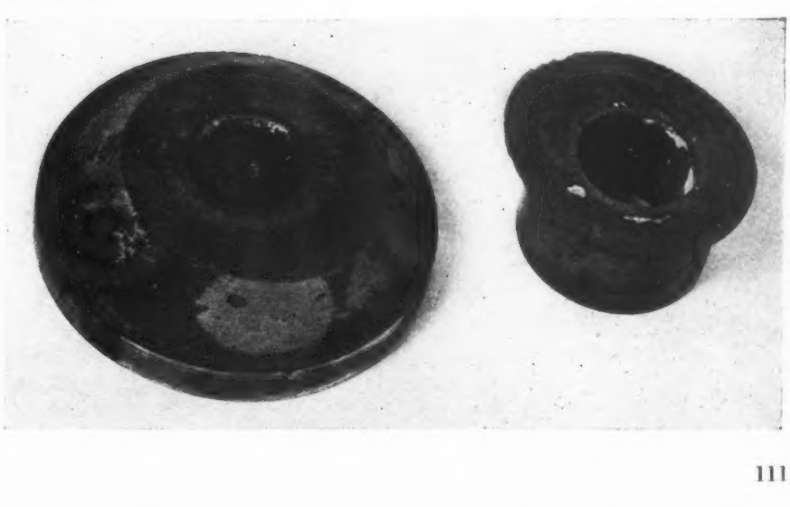


Fig. 99—Right—SAE 1095 cracked in water quenching. Rim transformed to martensite, flange did not, pulling the two sections apart. Alloyed steel should have been used, with an oil quench, to insure uniform hardening

element which will dissolve in austenite favorably affects the hardenability, assisting the steel to use its inherent capacity for hardness and strength due to its carbon content. In this respect carbon itself acts as an alloying element, improving the hardenability.

The value of alloys in increasing hardenability may be appreciated by consideration of the so-called "mass effect". Since the temperature of the outside of a quenched bar will have to drop before heat will flow from the inside, the interior will not cool as rapidly as the surface. As a result it will be more difficult for the center to form martensite than the outside. Moreover, the lag in center cooling rate will increase with increase in size. The effect can be demonstrated by a construction of temperature gradients across a bar at successive time intervals.

Obtaining data for such curves from actual quenches is exceedingly difficult. However, a qualitative picture may be obtained from consideration of an idealized condition. The "ideal quench" would be one having an infinite rate of heat tranfer. That is, the surface would be brought to the temperature of the quench medium instantaneously and would be held at that temperature, a condition fairly well simulated by a vigorous water quench. It will also be necessary to assume a constant specific heat, conductivity and density for the bar and to ignore the heat of transformation. For these calculations they were chosen at .1/cal./g./degree C., and .19 cal./sec/cm^2/degree C/cm. and 7.8 g/cm^3, respectively. To further simplify it will be assumed that cooling is from two faces only of a rectangular shaped bar, as shown in *Fig.* 95, a condition approached by the center of a large plate (4).

Alloy Aids Through-Hardening

Temperature gradients across a 1-inch thick plate for time intervals of ½, 1, 2 and 4 seconds are shown in *Fig.* 96. The "gate" of the transformation curve is passed in one-half second by a point .31-inch from the center, whereas it requires one second for a point .23-inch out, and about 1.9 seconds for the center. If this plate were of the eutectoid carbon steel whose isothermal curves are shown in *Figs.* 28 to 30§, the gate at 1000 degrees must be reached in ⅞-second after leaving the austenite equilibrium temperature, 1335 degrees. From *Fig.* 96 it may be seen that the outer 5/16-inch does this, leaving a core of increasing pearlite content ⅜-inch diameter. If, however, an alloy were added to increase the gate to a little over one second, the plate would be hardened through.

The condition for a 4-inch thick plate is illustrated by *Fig.* 97. The carbon steel now hardens less than .3-inch deep, and the alloy which would harden the 1-inch plate through only hardens the 4-inch plate little more than 5/16-inch deep. It is necessary to widen the gate to some 15 seconds to harden through the heavier plate.

It should be appreciated that this ideal quench can never be met fully. Heat transfer at the surface is not instantaneous, nor is the heat carried away instantly, the layer of quenching medium surrounding the part rising in temperature. Thus, while in the ideal quench the surface will always be hardened, in actual operations this may not be true. This departure from ideal conditions is best demonstrated by constructing hardness curves across actual bars quenched under operating conditions. *Fig.* 98 shows such gradients across various sizes of .40 per cent C rounds, data from *U. S. S. Carilloy Steels.* The variations in the hardness reflect the structures obtained from decreasing cooling rates—full martensite, mixed martensite and pearlite, and pearlite of successively coarser grades (1 and 5).

The same story may be told by holding size and material constant and decreasing the severity of the quench (6, 7 and 8). This is exactly what occurs in the Jominy

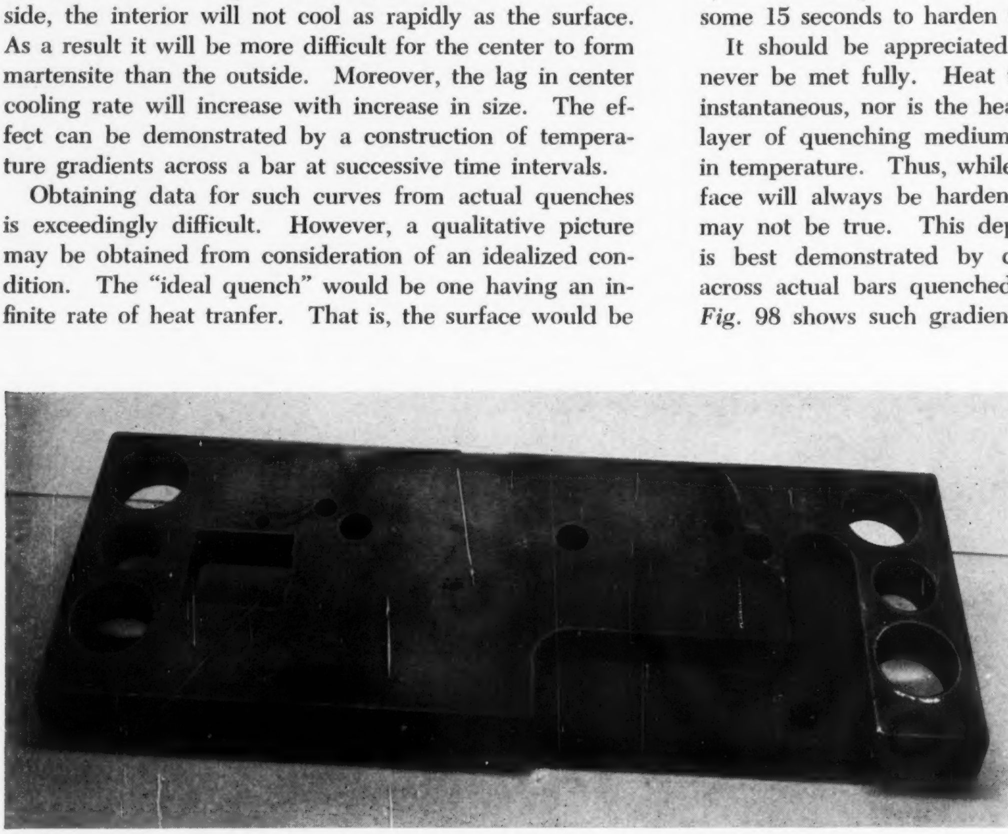
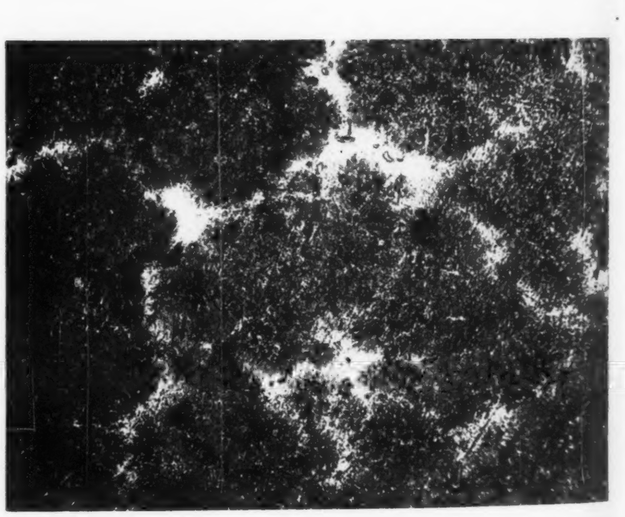


***Fig. 100**—Left—Carbon tool steel die plate cracked in tempering after water quench. Combination of light and heavy sections, low hardenability, caused non-uniform hardening*

***Fig. 101** — Below — Photomicrograph of failed SAE 4140 casting. Structure tempered martensite with ferrite network. Grain size about 2. Magnification, 500 diameters*



§MACHINE DESIGN, Nov., 1942, P. 82.

end-quench, *Fig.* 94. It is necessary, for comparable results, that conditions of quench be identical. A scale or carbon soot blanket will lower the quenching rate, as will also extreme variations in temperature of quench, particularly with aqueous solutions, rate of flow, temperature from which bar is quenched, etc.,(1).

Temperature gradient curves of *Figs.* 96 and 97 also help to an understanding of quenching stresses, cracking and warpage. The transformation to martensite is accompanied by an increase in volume. When the outside transforms the inside is still weak and yields to its pull, obtaining a fair degree of stress equilibrium. However, when the inside transforms, considerably later, the outside is hard and relatively brittle martensite, not nearly as well able to stand the pressure of the increased volume. The effect is to place the outside shell under internal pressure, developing high tensile stresses. It is for this reason that quench cracks occur near the end of the quench, or in the beginning of the tempering operation.

The ideal would be to bring all parts down in temperature uniformly, so that transformation occurred at the same time. This calls for the opposite of the requirements of hardening, a quench having a very low rate of heat transfer. For this reason the quench employed should be as mild as the hardenability will permit. It should be noted that the same results could be obtained by a *low* hardenability, one so low that the center transformed to pearlite while the outside went to martensite. It thus may be seen that under some conditions too high a hardenability may be obtained and under others, the reverse. This is demonstrated in the quench cracked parts of *Figs.* 99 and 100.

It may now be seen that the prime purpose of the addition of alloying elements to steel is either to permit securing martensitic structure from a given quench, with an increased size, or reducing the severity of the quench. If the largest section to be hardened were only about ¾ to 1 inch, straight carbon steels would serve quite well. It is when the larger sections are employed that the value of alloys is felt(9, 10).

Should Dissolve for Hardening

To secure the full value of alloys for increasing hardenability it is necessary that they be fully dissolved in the austenite previous to quenching. Considerable sluggishness is exhibited by many of the elements either in going into solution or in permitting other carbides to dissolve and diffuse to a homogeneous disposition. While plain carbon steels will be ready to quench in a matter of minutes at temperature the alloys may require higher heating and far longer holding. The extent of this will depend not only upon the alloy, but also upon its distribution at the time of heating(11). The disastrous effects of ignoring this factor are illustrated in *Figs.* 100 and 101. Not only is the network structure brittle, but the nonuniform hardening resulted in high quenching stresses. Some castings actually failed in the quench, nearly all others failed in service. Alloy castings in particular should be homogenized at high temperatures so that the elements may be picked up by the austenite and diffused.

All the alloys retard the softening in tempering, both as regards time and temperature. This undoubtedly accounts for the creep resistance of the high alloyed steels (1 and 3). The time effect extends over a long interval, commercial practice of tempering from ½ to 2 hours giving substantially constant results.

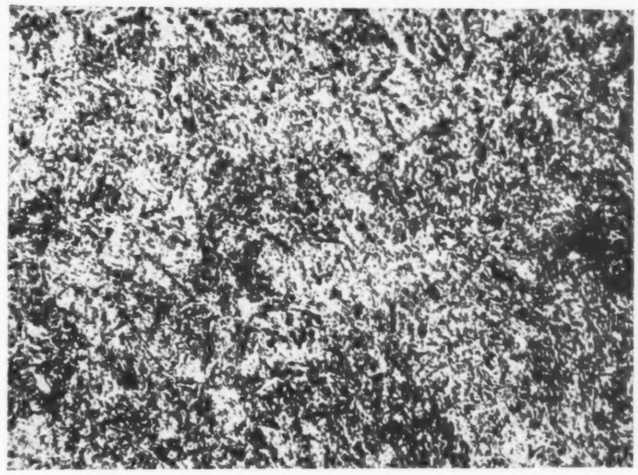


Fig. 102—Photomicrograph of properly treated casting of Fig. 101. Structure is tempered martensite

Steels do not soften alike with temperature. The addition of alloying elements calls for a higher tempering temperature to obtain the same hardness. In general, those alloys that are the most sluggish in going into solution in the austenite most strongly resist the tempering (12 and 13).

There is amazingly little to choose between steels heat treated and tempered to the same hardness. Janitsky and Baeyertz(14) investigated alloy steels varying in carbon from .3 to .45 per cent, both water and oil quenched. *Fig.* 103 is reproduced from their data. The greater variation in values at high hardnesses may well be explained by the lower reliability of such data. Quenching stresses, notches, etc., are of much greater import.

There are some applications where the use of alloys may be advisable, even if a plain carbon steel will harden out. The use of a milder quench with its attendant lower residual stresses, and the employment of higher tempering temperatures with a consequent further reduction in these residuals, may at times be of considerable significance. With these and a few other exceptions, however, the choice of which alloy—assuming that sufficient hardenability has been obtained—may be dictated by other requirements such as economy, availability, machinability(15 and 16), rather than on unwarranted assumptions of phenomenal qualitites imparted by some particular element.

Special Applications Distinguish Alloys

As applied to special applications more can be found to distinguish one alloy from another. Low temperature impact is improved by the ferrite-soluble elements, nickel being outstanding in this regard. High-temperature creep is aided by the strong carbide formers, molybdenum being outstanding in this regard. Abrasion again is aided by hard complex carbide particles such as those formed from chromium, vanadium, tungsten and molybdenum although, in general, only the first will be present in sufficient quantities in constructional alloy steels to be of

much value. Atmospheric corrosion resistance depends upon the formation of a tight oxide, which acts then to prevent further corrosion. In large amounts chromium is of value, as witness the 12 per cent grades of stainless cutlery steels, Type AISI 430.

Of the lesser effects of alloying elements, grain size has received a great deal of attention, deservedly so before the days of grain-size control, unwarrantedly today. It is for this reason that so much credit was given to the toughness of vanadium steels with their large amount of rather insoluble carbides. Aluminum has now supplanted such applications.

All the alloys lower to some extent the carbon content necessary for the eutectoid composition. This is true even of the ferrite soluble elements. The effect is of little importance in the carbon range employed in through-hardening constructional steels. It is of some importance, however, in carburizing. By going further beyond the eutectoid composition the risk of a hypereutectoid deposition of carbide with its attendant embrittling effect is increased.

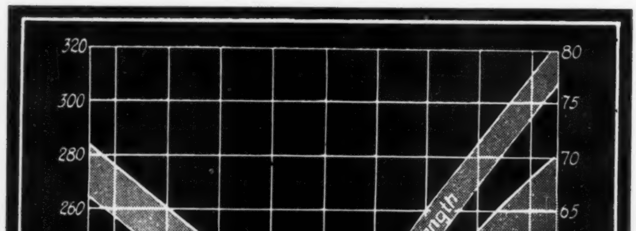


Fig. 103—Variations in physical properties of a number of SAE alloy steels, varying in carbon from .3 to .45 per cent. Heat treated to the same hardness

With the exception of the free-machining steels, attained by manganese sulphide or other inclusions such as lead to break the chips, machinability is reduced by alloying additions. This is an extremely difficult quality to correlate. So much depends upon the structure, whether the carbide is present in coarse or finely divided plates, or in a spherodized condition. Also, an apparently minor alteration in clearance or rake angle of a cutting tool, or perhaps a change to a different cutting fluid or a different rate of cut and so forth may completely alter this quality. Full annealing is not always the best condition. A case at hand in the writers' experience was that of a number of .45 per cent C steel forgings in which annealing only accentuated the trouble. The cure was a normalized structure of finely divided pearlite, attained by air cooling from over the critical, followed by a drawback at 1200 degrees.

The next article will summarize the effect of the addition of alloying elements to steel for each alloying element, with stress placed upon those elements or combination of elements of steels employed in constructional practice and available generally in warehouse stocks.

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Standard for Engineering Graphs Approved

PRESENTATION of engineering data in graphical form demands careful attention in order that the graph serve its purpose of promptly conveying significant information to the reader. In connection with engineering reports prepared for executives, clear presentation is of particular importance. On this subject a helpful guide has recently been approved by the American Standards association and issued in the form of a 28-page booklet, "Engineering and Scientific Graphs for Publications."

Choice of scales and their designation, coordinate rulings, curves, plotted points and titles are discussed and illustrated in Part 1, the weaknesses of poor style being pointed out by examples. Part 2, covering construction of original graphs to be used for reproduction, is particularly concerned with appearance factors and the effect of size reductions on line thickness, lettering, and symbols for plotted points.

The new standard, which is designated ASA Z15.3—1943, is sponsored by the American Society of Mechanical engineers from which copies may be obtained.

Engineering Aspects of Plywood

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BECAUSE of the great potentialities of plywood in future designs, strength characteristics of this material and design features of typical plywood aircraft members should be particularly interesting to design engineers. The accompanying article is based on a paper delivered before the National Aeronautic meeting of the S.A.E. by the author who is chairman of the ANC executive technical subcommittee on wood for aircraft

PRESENT efforts to turn out wood aircraft in mass quantities and at the same time take all advantage of recent developments in glues, fabrication technique and design, has brought us face to face with many new problems as well as some skeletons of the past. Although it is not possible to give the answer to any of these problems with either the degree of thoroughness or accuracy that is desired, a partial answer at the present time may be of more value than the complete answer at a later date.

It is often not understood just why wood strength tables fail to give values for tensile strength, although the tensile strength of wood parallel to the grain is very high, being about 18,000 pounds per square inch for straight-grained spruce. Thus, the ratio of tensile strength to specific gravity for spruce is approximately 45,000 as compared to 23,600 for 18-8 stainless steel in the full-hard condition.

The reasons why this high tensile strength cannot be utilized in structures are (1) the low shear strength of wood and (2) the highly critical effect of stress concentrations and slope of grain on tensile strength. It is practically impossible to obtain pure tension stresses in a member, or to devise fittings that will permit the tensile strength of the full cross section to be developed. As a result, shear failures result rather than tension failures. Since the permissible slope of grain in aircraft lumber is 1 in 15, the allowable tensile strength must be set accordingly. For spruce, a grain slope of 1 in 15 will reduce the straight-grain tensile strength to about 9,400 pounds per square inch. A similar reduction was also found to be true for other species and, because these tensile strengths (for a 1 in 15 grain slope) checked the modulus of rupture in static bending closely, the latter property has been conveniently used to represent the allowable tensile strengths parallel to the grain. In contrast to the high tensile strength of Sitka spruce parallel to the grain, the specific strength in tension perpendicular to the grain is only 600, and in shear parallel to the grain, 2500, as compared to 23,000 and 16,000 respectively for 18-8 full-hard stainless steel. (These values are approximate.)

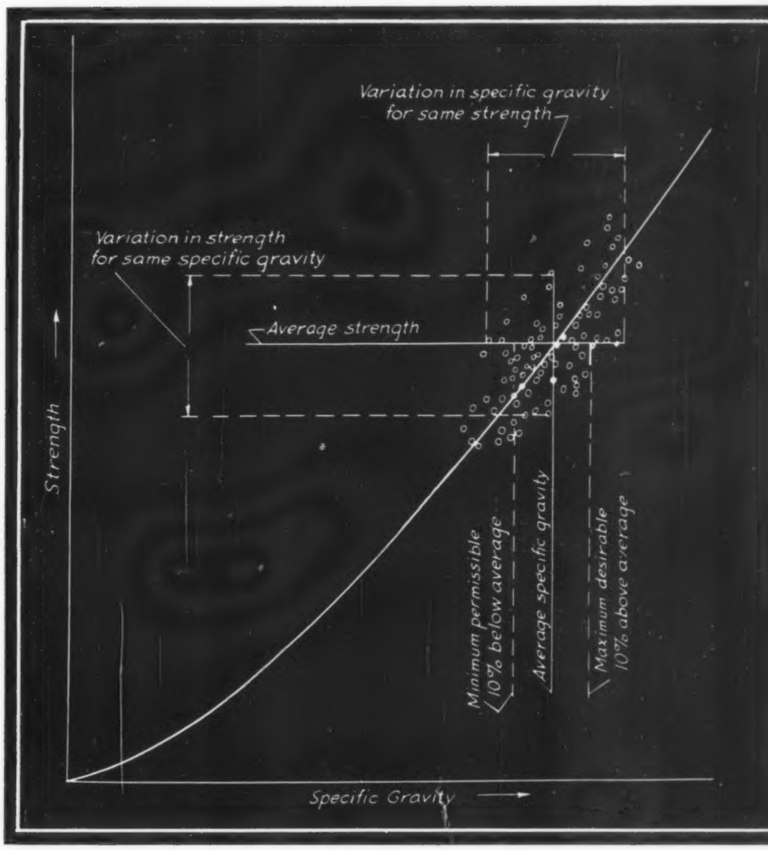


Fig. 1—Variation in wood strength with specific gravity

Strength properties of wood are much affected by changes in moisture content and, unfortunately, the moisture equilibrium of a piece of wood will vary with the conditions of relative humidity and temperature to which it is exposed. Near Seattle, Wash., and other humid regions the wood moisture content averages approximately 15 per cent, while in certain regions of the Southwest a moisture content of five per cent is not un-

common. The differences in strength of Sitka spruce for these two moisture contents are shown in the accompanying table.

From the standpoint of efficiency and avoidance of shrinking and swelling difficulties, it would be desirable if wood aircraft could be designed for operation in a particular geographical location. For example, a spruce wing spar of 1 by 6-inch cross section designed on the basis of 15 per cent moisture content could be reduced to slightly less than a ¾ by 6-inch cross section on the basis of a 7 per cent moisture content, with a resulting weight saving of 28 per cent.

Glues: A number of thermoplastic resin glues have been developed which appear to have all of the desirable characteristics of good bag-molding or final-assembly glues, but these glues should not be used in highly stressed parts of the structure. Thermoplastic glues are subject to "creep" at elevated temperatures (approximately 160 degrees Fahr.). Since wood surfaces coated with a dark camouflage paint may reach temperatures well over 200 degrees Fahr. in desert regions, separation of the glue joints is likely to occur.

Glue manufacturers in general will not recommend nor guarantee their glues for use in gluing dense materials, such as compreg or plastic laminates, and it becomes necessary for aircraft manufacturers to rely upon rather meager data resulting from their own experience with certain types of glue. A few sources of information, however, indicate that reasonably good results can be obtained in gluing compreg to compreg or plastic laminate to plastic laminate if a low-temperature phenolic-resin glue is used and the surfaces are roughened slightly.

When compreg is glued to normal wood it is recommended that a layer of impreg be inserted between the two materials. The primary objective is to reduce as much as possible the stress concentration in any single glue line due to pronounced differences in moduli of elasticity.

Specific-Gravity Corrections: Unfortunately the term "minimum guaranteed properties" is not applicable to wood and plywood structures and, as a result, the current attempts at correction of static-test data have led to nothing but confusion. It is true that a "minimum permitted" specific-gravity value is specified for all aircraft lumber but the design allowables for these species are related to "average" specific-gravity values rather than to this "minimum permitted." As shown in *Fig.* 1, the magnitude of the average specific-gravity has been determined by a consideration of values both above and below this so-called average. It is thus apparent that any strength property for a given species should be associated with a "range" of specific-gravity values rather than an average, and that both the upper and lower limits of this range should be specified in any wood strength table. Since the minimum permitted value was placed ten per

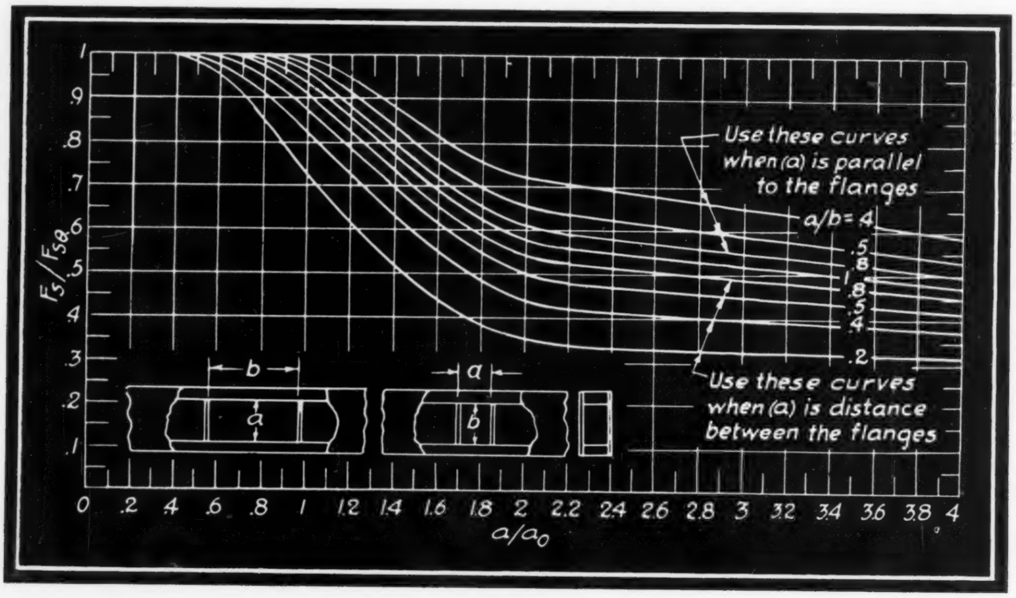
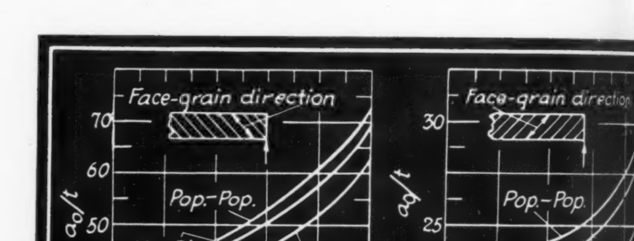


Fig. 2—Left—Allowable shear curves for plywood webs. Dimension (a) is chosen so that it is less than (b)



Face Grain in Compression **Face Grain in Tension**

Fig. 3—Below—Curves for value of a_0, for determining allowable stress in Fig. 2

Sitka Spruce Strength Properties

Strength property	Average strength at 7% moisture content (psi)	Average strength at 15% moisture content (psi)	Per cent increase in strength from 15% to 7% moisture content
Static bending proportional limit	8,950	6,200	45
Static bending modulus of rupture	12,800	9,400	36
Maximum compression perpendicular to grain	1,180	840	40
Maximum compression parallel to grain	7,600	5,000	51
Shear strength parallel to grain	920	750	23

cent below the average, it is reasonable arbitrarily to set a maximum value at ten per cent above the average. Thus, for spruce the specific gravity range should be .36 to .44. This upper limit need not be considered as the maximum permitted but rather as the maximum desired in order to avoid excessive structural weight.

Another point that should be noted in *Fig.* 1 is the considerable variability in wood strength properties for identical values of specific gravity. This variation is typical of all of the wood-strength and elastic properties, some to a greater degree than others, but the overall average of this variation is approximately ten per cent.

In view of the above points it is believed that static-test data for isolated wood structures, either major assemblies or component parts, should not be corrected for variation in specific gravity. In connection with this policy, however, the aircraft manufacturer should select material that has a specific gravity between plus or minus ten per cent of the average value specified in order to obtain results which are indicative of the average strengths.

Shear Webs for Box-Beams: In the past the design of box-beam plywood shear webs has been based upon two sources of information, National Advisory Committee for Aeronautics *Technical Report* No. 344 by G. W. Trayer, and Air Corps *Information Circular* No. 516 by R. W. Miller. Both of these reports were the results of static tests. Because of the differences in test conditions, however, the quantitative design values recommended in each case were not in agreement. In some instances this discrepancy mounted to several hundred per cent, showing the need for more satisfactory design information and clarification.

In connection with the current Army-Navy-Civil research program the Forest Products Laboratory in Madison, Wisconsin, made a study of all available test data on the strength of plywood shear webs. Further study and experimental work has produced the set of curves shown in *Fig.* 2 for the design of plywood shear webs in I-beams and box-beams.

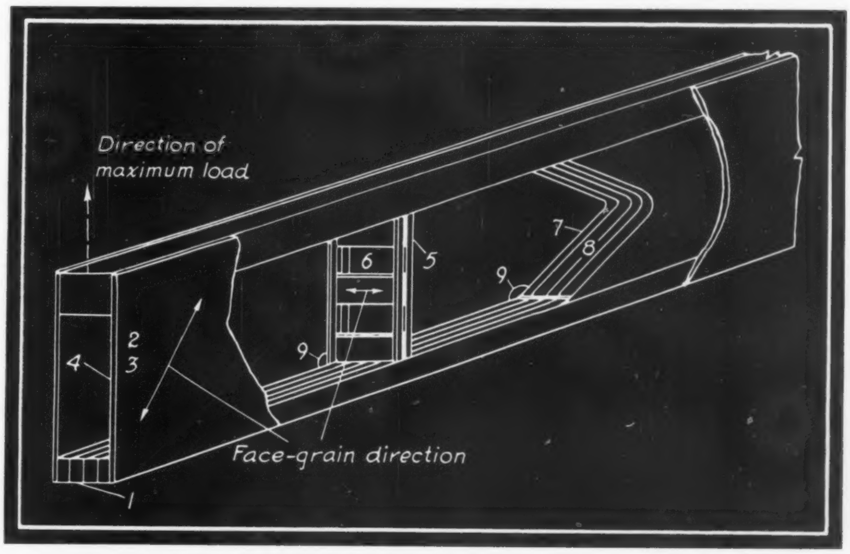
To compute the allowable shear stress F_s for a plywood web of three-ply, 1:2:1 construction direction of stresses relative to face-grain direction is determined as shown by the sketches in *Fig.* 3. Then a_0/t is obtained from the appropriate set of curves[1] in *Fig.* 3, where t is the web thickness. From this a/a_0 or the abscissa in *Fig.* 2 is obtained from which $F_s/F_{s\theta}$ is obtained, where $F_{s\theta}$ is for 45-degree birch-birch, mahogany-mahogany and poplar-poplar 3420, 3020, and 2270 pounds per square inch, respectively[2].

Although strict interpretation of the curves in *Fig.* 2 apply only to plywood beam webs, it is believed that they may also be used to calculate the shear strength of other types of plywood shear panels (such as in wing skins or fuselage coverings that have little or no curvature) provided certain precautions are taken. If any edge of a panel is not rigidly restrained against movement in its own plane, the lower set of a/b curves should be used. An example of this might be a plywood panel in the wing covering at the inboard end of an outer panel where the end rib does not afford a rigid spanwise restraint to the edge of the panel.

Determining Shear Strength

The shear strength of a panel which is rigidly restrained along all edges in its own plane may be determined by use of the upper set of a/b curves. A panel whose edges are entirely within a larger plywood sheet, or a panel whose edges are restrained on one or more sides by a heavy member and on all other sides by a continuation of the plywood, will fall into this group. It should be noted that the curves in *Fig.* 2 are applicable regardless

Fig. 4—Detail design features of box spar with explanatory notes



1. **Tension flange laminated to avoid effects of compression failures not found in inspection of material**
2. **Web face grain 45 degrees to spar axis for greatest shear strength**
3. **Web face grain in compression under critical loading for greatest buckling strength**
4. **For 50-50 construction, most efficient plywood web material is three-ply**
5. **Diaphragm verticles fitted snugly between flanges and dimensioned so as to provide large glue area for web attachment**
6. **Diaphragm cross members of plywood, flexible enough to yield slightly when web buckles**
7. **Filler block tapered to avoid stress concentrations. If designed to transmit loads, taper ratio should equal ratio of maximum tension stress in adjacent flange to horizontal shear strength of filler block**
8. **Filler block cross-banded to prevent checking and other difficulties resulting from swelling and shrinking due to moisture changes**
9. **If water has access to interior, ¼-inch diameter (minimum) drain holes should be made in each compartment flush with lower flanges Similar vent holes should be placed in web near upper flange.**

[1] Values for a_0 may also be calculated from Equation 2:45 in ANC Wood Handbook (restricted) for other constructions.

[2] Other values of $F_{s\theta}$ may be calculated for other plywood constructions from Equations 2:30 and 2:31 in ANC Wood Handbook (restricted).

of the direction of the plywood face grain; the variation in shear strength is compensated for in the terms F_{so} and a_o.

Although it is common practice to design metal shear webs in the tension-field range, this same practice is not advisable for an efficient design of plywood webs. It has been found by test that when the face-grain direction is parallel to the tensile stresses (this would be the direction of greatest tensile strength for plywood having equal veneer thicknesses) the buckles become large and the resulting bending stresses reduce the ultimate tensile strength far below that normally expected. If the face-grain direction is run perpendicular to the tensile stresses the resistance to buckling is increased, but in this case the tensile strength of the plywood is low, even in the unbuckled state. Consequently there does not appear to be any "best way" of running the face-grain direction for plywood tension-field webs. Further, the tension-field buckles tend to project themselves into the glue joints, thus causing premature failure due to separation of the stiffeners and

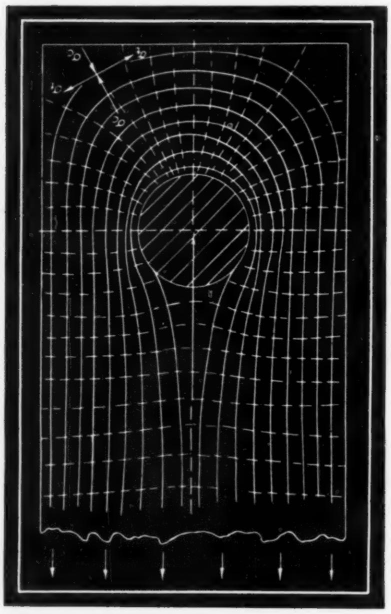


Fig. 5 — Principal stresses set up in isotropic plate by bolt load

flanges from the plywood webs. A number of points which should be given consideration in the design of box-beams have been noted in *Fig.* 4.

Efficiency of I-Beams: Box-beams seem to have replaced to a large extent the use of I-beams in aircraft construction, although the latter type is more efficient on a strength-weight basis. Referring to the curves of a_o/t vs. a/b in *Fig.* 3, a_o varies directly with t. For the same total plywood web thickness, therefore, the value of a_o for an I-beam would be twice that for a box-beam. Referring to *Fig.* 2, this means a value of a/a_o of one-half that for the corresponding box-beam with a very appreciable increase in the allowable shear stress over the normal range of design.

Another factor which should be considered is that of buckling of the plywood shear webs. In the case of the box-beams, any buckles which form under load become deeper as the load is increased and these tend to pull the web away from the flange. In the case of the I-beam, there is no opportunity for the web buckles to project on through the web-flange attachment.

Some arguments may be raised against the use of I-beams because of the difficulty in attaching ribs. Such arguments have little basis as it is simple to use web stiffeners as rib-attachment points, or it also is possible to devise various "mechanical shear joints" which take advantage of the I-beam cross-section profile. This indicates that designers would do well to consider the use of I-beams for plywood-covered wings and not follow along blindly in the use of the more-or-less standard box-beams.

Stiffeners for Compression Panels: An important problem in the design of reinforced plywood compression panels, such as wing and fuselage coverings, is to determine what stiffener width is required to provide sufficient glue area to prevent separation of the plywood and stiffener under load. This glue-area requirement appears to be primarily a function of stiffener spacing, plywood thickness, plywood construction, and face-grain direction.

Design information on this problem, either theoretical or experimental, is extremely meager and manufacturers are urged to conduct stiffened-panel compression tests to determine a suitable stiffener width rather than to assume some value arbitrarily. Such preliminary tests may save time and expense by avoiding extensive redesign if separation of the plywood and stiffeners causes failure.

Applying Rule of Thumb Data

Such test data as are available indicate that the following rule of thumb should be satisfactory for 45 degrees mahogany-poplar multi-ply plywood with Douglas fir stringers. When buckling of the plywood panel is expected before the ultimate design load is reached, the width of glue line attaching stringer to plywood should be approximately six times the plywood thickness. When buckling is not expected to occur below the ultimate design stress the width of glue line may be reduced. The glue-line width need be only three times the plywood thickness when the critical buckling stress is more than twice the ultimate design stress. Theoretically, a plywood panel will have the greatest tendency to separate from stringers at the ultimate design stress when the stress at which buckling first becomes noticeable in the actual structure. It should be noted that this stress may be only 50 per cent of the theoretical calculated value, due possibly to initial eccentricities.

Bearing Plates — There is a serious problem in the design of bolted joints for solid-wood members under tension loadings because of the tendency for wood to split and shear out in front of the load. One of the most common recommendations for good design practice in wood-aircraft construction is for the use of plywood bearing plates for all bolted connections. It is unfortunate that plywood plates have been termed "bearing plates" as this has led many to believe that plywood itself has a high-bearing strength which is not so. Actually, these plates should be called "reinforcing plates" or some such similar term since their primary function is to prevent splitting of the solid wood members to which they are glued. A basic understanding of this point can be gained by referring to *Fig.* 5 which shows the results of photoelastic studies of a bolted joint in an isotropic material. (Although wood is anisotropic, it is believed that this figure is at least qualitatively correct when applied to a wood

member.) The solid lines represent the lines of principal tensile stress while the dotted lines represent the companion principal stress.

Since wood is weak in tension perpendicular to the grain, it is obvious that the tensile stresses shown will cause the member to split open at low loads. If, however, a piece of plywood is glued to each bearing face of a bolted member the plywood will resist the stresses tending to cause cleavage, thus enabling the solid wood to develop a much greater shear strength before failure.

It can also be seen from the stress trajectories in *Fig.* 5 that the direction of maximum shear stress just below the bolt is essentially parallel to the direction of loading. For this reason plywood reinforcing plates should be laid so that the face-grain direction runs 45 degrees to the direction of loading.

For the same overall thickness it has been found by test that a plywood-faced bolted member may develop as much as 50 per cent more strength than will the same member without plates. The thickness of plywood to be used is necessarily a function of the thickness of the bolted wood member. It is recommended that a minimum thickness of ⅛-inch 45 degree plywood be used on each face for a total thickness up to two inches and varying linearly thereafter to ¼-inch for a total thickness of five inches. The plywood should be 50-50 construction.

It is further recommended that mahogany-poplar plywood be used for reinforcing low-strength woods, such as spruce and poplar, and that birch-birch plywood be used for reinforcing high-strength woods such as birch, maple, hickory, and walnut. Composite members made in this manner will give bearing strengths almost as high as would be obtained from the use of "impreg" or "compreg."

High and Low-Density Woods in Combination: Although it is entirely possible to use high and low-density woods in combination in highly stressed members satisfactorily, the differences in elastic properties are a potential source of trouble. In certain aircraft structures, particularly wings, premature failures in static tests have sometimes been caused by the elastic incompatibility of high and low-density woods.

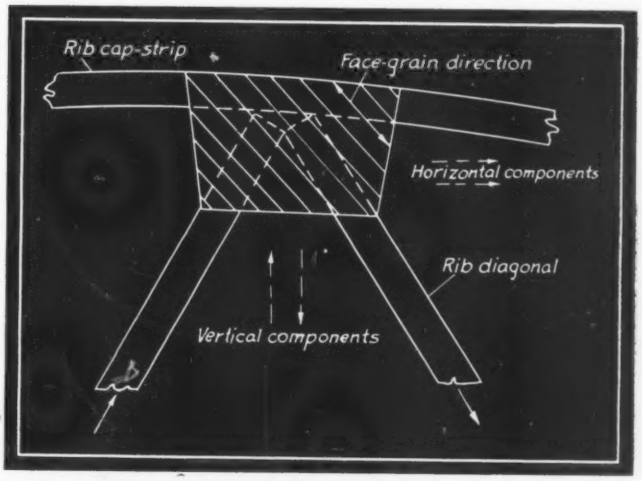
Indicate Design Changes

A specific example of an incompatibility failure is one which occurred on a wing with hickory spar caps and square-laid poplar plywood covering (face-grain direction running spanwise). In the stress analysis of this particular structure the skin was assumed to carry no bending stresses. Nevertheless, tension failures occurred in the skin at about 80 per cent of the design load. A review of the stress-strain characteristics of the two woods revealed that while the modulus of elasticity of poplar was much less (hence the corresponding stress for a unit strain was less) the total work of which the hickory was capable of doing under load was so much greater that the ultimate strength of the poplar was exceeded while the hickory was yet well within the elastic range.

The remedy for such an occurrence is to reduce the modulus of elasticity of the plywood covering by rotating the face grain to be 45 degrees with the spanwise direction. This latter arrangement is particularly desirable in view of the fact that the original assumptions in the above-mentioned stress analysis are more nearly fulfilled and the torsional properties of the wing are improved.

Leading-Edge Structures: An interesting case of leading-edge failure on a conventional two-spar, trussed-rib, fabric-covered wing (plywood nose) was observed during static tests. In the design of this particular wing, the plywood leading-edge cover had been disregarded structurally, even though it was securely fastened to the ribs and the front spar. As the wing deflected under load the leading-edge plywood cover was stressed accordingly. Failures began to occur at as low as 40 per cent of the design load in the plywood nose cover in the vicinity of access cut-outs, and finally the entire leading-edge structure was torn from the wing at less than 80 per cent of the design load. Designers believe they are being conservative by neglecting the load-carrying capacity of certain parts whereas, in reality, they are inviting trouble. A failure or series of failures, caused by structural members which are arbitrarily considered nonstructural, are just as costly and even more irksome than failures in members which were given sufficient consideration.

A general weakness of leading edges in stressed-skin



Plywood gusset should be 50-50 construction and not less than three plies
Outer plies of gusset and cap-strip material should be of about the same density
Plywood face grain should be 45 degrees to cap-strip, and run in direction of tension load.

Fig. 6—Design of typical trussed-rib plywood gusset

wood structures appears to be the tension attachment of ribs to spars. It is fairly easy, in the usual case, to obtain adequate shear connections with corner blocks or plywood angles but tension connections are affected by skin stresses from sources other than the direct rib loads. This fact, coupled with the inherent difficulties involved in making a satisfactory tension connection in wood structures has caused considerable leading-edge trouble.

Gussets for Trussed-Type Ribs: Little or no attention is given to the design of plywood gussets for use in attaching diagonals to cap-strips in the conventional truss-type rib. This oversight usually results in premature failures in rib static tests and consequent delay while the "gusset trouble" is remedied. The most efficient type of plywood gusset for a particular location will depend upon the rela-

(*Concluded on Page* 208)

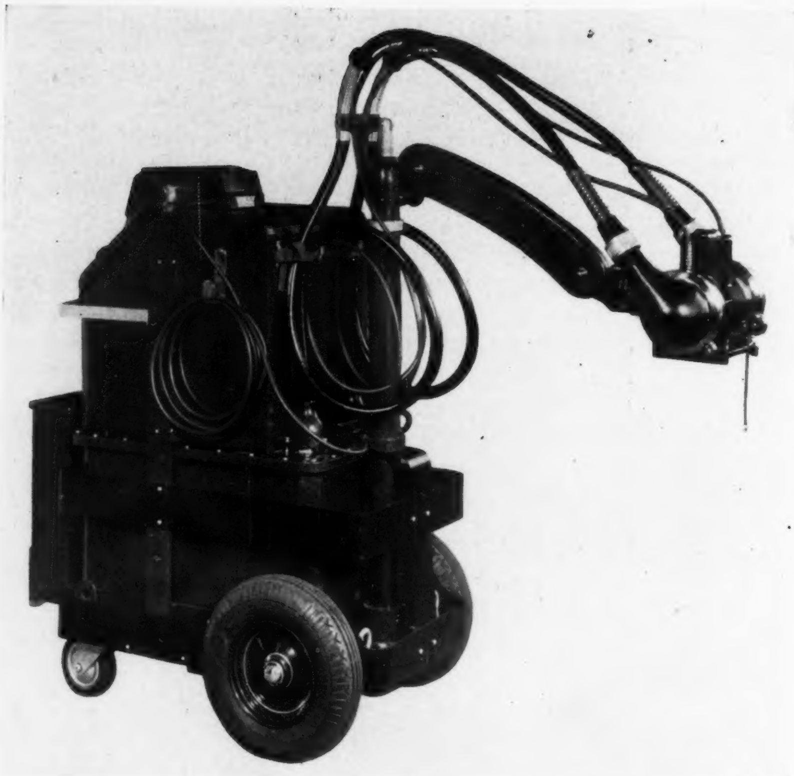


Fig. 1—Above—Pneumatic wheels protect X-ray unit from shock. Also they prevent distorted images by damping floor vibrations

Fig. 2—Below—Portable vibrating screen carries major load on large pneumatic-tired wheels. Clevis yoke provides for steering

Selecti

By John W. Greve

UPON satisfactory functioning of its wheels or casters often depends the acceptance of a mobile machine. For each design, however, special consideration is necessary to obtain the best wheels or casters for the particular job for which the unit is intended. Selection of these parts is so important that it should be common practice to make the selection at the outset of the design.

There are certain general rules which apply to the use of wheels—the primary subject of this article. They must, in the first place, assure the user a number of years of dependable service. Although usually considered minor items, they can be such an annoyance as to influence user's goodwill even though the rest of the unit functions satisfactorily. Antifriction bearings are desirable for both the swivel and wheel bearings, particularly if medium-weight or heavy-weight machines are to be moved frequently. A balance between a high degree of maneuverability, quietness and ease of operation is essential. A good compromise for small, manually-propelled machines is to make the plain, nonswiveling wheel soft and the swivel casters hard.

Should Provide Adequate Wheels

Many otherwise excellent designs of small machines have overlooked or not provided adequately for the rolling equipment. For instance, a case at point is a relatively light, well-known machine having an enclosed, styled housing covering the machine to the floor. When casters were applied, however, it was impossible due to a relatively high center of gravity to use casters of sufficient diameter to provide easy movement over minor bumps in the floor. The offset of the swivel for clearance of the enclosing housing rendered the machine unstable when the casters were turned to the inward position. This oversight in design rendered necessary the selection of a too-small diameter wheel to maintain as much stability as possible.

For the sake of wheel selection, ma-

Wheels and Casters

Part I

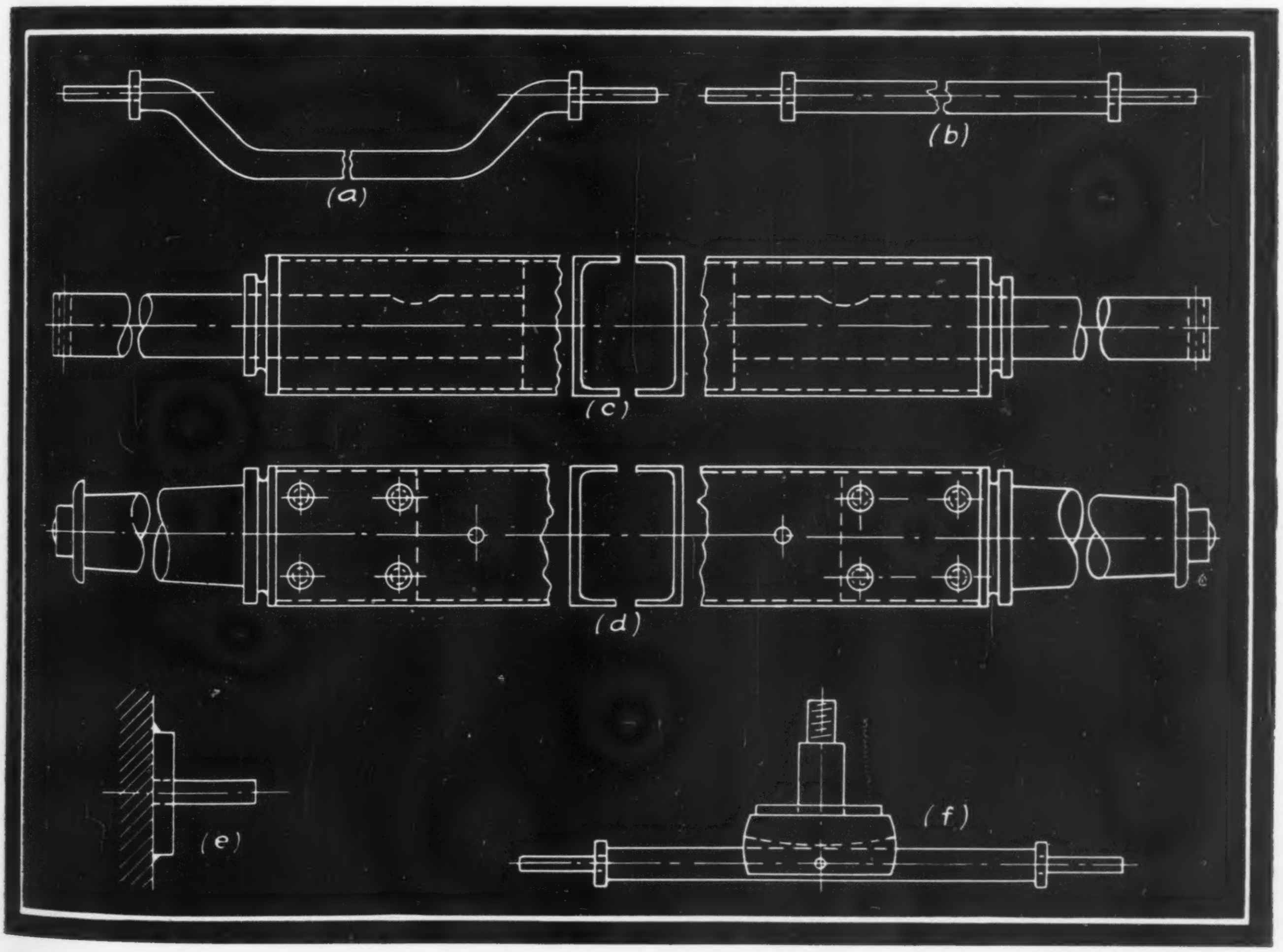
chines are grouped into five general classifications:

1. Manually propelled, involving speeds up to three miles per hour
2. Industrial truck or tractor drawn, involving speeds up to ten miles per hour
3. Highway tractor drawn, involving speeds up to twenty-five miles per hour
4. Automotive towed, involving speeds up to sixty miles per hour
5. Self-propelled, involving any speed range.

These groups are so selected because speed of operation is usually the primary governing factor in the design of wheels and their supporting members. For instance, shock loads or side thrust are relatively unimportant factors in the first group. They become of major importance, however, in the last three groups. Added to these problems for the fifth group is the necessity for the power wheels to transmit torque and provide sufficient traction. Because considerable attention has been given the latter groups, and because design becomes more highly specialized, primary attention will be directed in this article toward the first and second groups.

With respect to selecting wheels for equipment operating at speeds in excess of ten miles per hour, it would be relatively simple for wheel manufacturers to provide wheels if pneumatic tires were always available. Being highly restricted at present, it is necessary to make application to the government for each specific use. As a result steel wheels are being utilized and it is being found that these will not stand up when drawn over bumps such as tracks, etc. Shock loads become too great and no wheels have been designed to date that will withstand the severe service. Something, no doubt, will be done in the near future to relieve the more severe and critical situations in war plants particularly. Meanwhile, it is necessary to design for the best compromise possible

Fig. 3—Typical axles for wheel mounting. At (a) is drop-type axle, at (b) straight bar axle, (c) and (d) built-up axle employing channel frame and stub shafts, (e) stub axle, (f) clevis yoke for swivel axle



and provide for wheel replacements.

After the primary consideration of speed with its attendant problems is determined, the major factors remaining are:

1. Height of center of gravity relative to wheel base
2. Turning radius required for maneuverability
3. Stability on extreme turn
4. Floor or surface conditions
5. Load range, stable or shifting type
6. Value and sensitivity of equipment.

Each of the foregoing factors is directly affected by the speed of operation. Some, also, are interdependent upon each other. Center of gravity must be kept sufficiently low and wheel base sufficiently large so that no tendency to overturn is approached even at extreme turns. Turning radius is usually a compromise giving as much maneuverability as is consistent with the size of the unit. If

Fig. 4—Small cast wheels provide mobility within working area for welder unit

operation in closer quarters is desired, redesign is indicated to provide a more compact unit, lower center of gravity, or a smaller wheel base. In some instances, it is possible to utilize a smaller wheel or a modification in wheel suspension.

Conditions of floor dictate the minimum size of wheel that may be used and also the width of wheel face. The larger the wheel, the easier the unit can be moved and, when relatively long distances or frequent moving is required, wheels as large as possible are selected consistent with the other design factors involved. There is no arbitrary standard governing the selection of wheel size. Width of face should be sufficient to withstand the loads, to prevent marring of floors and to straddle cracks.

If load is fixed it is possible to distribute weight proportionately over the wheels, carrying the major load on the larger nonswivel wheels so that the smaller swivel wheels or casters may turn more easily. If the load is "shifting" such as encountered with liquids, weight distribution cannot be apportioned to each wheel.

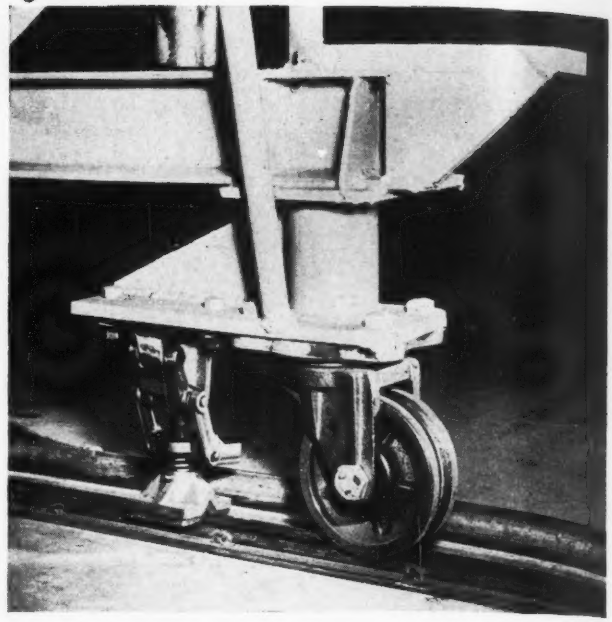


Fig. 5—Grooved caster wheels running on angle-iron tracks in the form of inverted V-track obviate trouble from chips and cotter pins falling on track

Proper cushioning and protection from shock is necessary for instruments and other mechanisms, the provisions depending upon their sensitivity and value. Pneumatic tires and spring suspension are the usual and simplest methods utilized. Often delicate mechanisms may be isolated separately in a subassembly.

To dampen floor vibrations and prevent them reaching the X-ray unit in *Fig.* 1, 16-inch pneumatic wheels are employed. Otherwise distortion of the X-ray image may result. The tires also protect the equipment from shock when being moved from one location to another. A 6-inch industrial plate type caster swivel is used at the rear

Fig. 6—Below—Industrial truck utilizes disk wheels and solid rubber tires

of the machine to allow steering. Face of this caster is sufficiently wide to prevent dropping into cracks.

Vibration isolation of an opposite nature is obtained from the rubber tires on the vibrating screen in *Fig.* 2. In this application the shaking of the screen itself is dampened and prevented from reaching the floor.

Many designs of axles and wheel mountings may be employed to fit a certain set of requirements. A few types, typical of the wide variety that may be utilized, are shown in *Fig.* 3. Simplest of these is the straight axle, *Fig. 3b*, usually fastened to the machine frame with U-bolts, drilled brackets or through a spring suspension. This type of axle is used in sizes up to four inches, from round or square stock. To provide for large wheels and yet keep the machine low, drop frames may be employed. Where it is uneconomical to offset the frame, a drop axle, *Fig. 3a*, is often used for sizes ranging from ¾ to 2½-inch.

Another type frequently employed, especially where relatively large wheels or a minimum clearance from the floor are desired, is the stub axle, *Fig. 3e*. Plate is bolted, riveted or welded to a frame member.

For heavy machines such as portable saw mills, well drillers, etc., axles may be built up of two channel sections with the axle running through the channels and supporting blocks welded to the channels; or stub axles may be riveted between the two channel members as shown in *Fig. 3d*. A variation of this latter method involves casting a block of steel upon a machined stub axle, *Fig. 3c*. The block is shaped for insertion between two channels and welded into position.

For steering assembly, automotive knuckle type is not used frequently for manual or power-drawn equipment, nor is the fifth wheel or conventional turntable axle employed except by wagon manufacturers. Instead, the usual portable mounting utilizes a clevis yoke for the swivel similar to *Fig. 3f*. A bronze thrust collar serves as the turntable bearing. Axle is mounted on a pin and the yoke is relieved at the sides to allow axle tilting.

Wheels are either built up or solid, depending upon size and service conditions. Built-up wheels may utilize a combination of stampings and castings as well as cast or stamped hubs with wire spokes assembled in a rolled rim. Cast wheels, *Fig.* 4, are limited as to size and, when large, become relatively heavy. Where thrust and shock loads are encountered, built-up wheels are indicated.

Rubber-tired wheels are available generally in three types, pneumatic, zero pressure and solid, *Fig.* 6, the selection depending primarily upon the amount of cushioning desired. Vulcanized fiber for wheels is highly restricted for war purposes. Its use has not as yet been clarified, and in view of the substitutions available it may not be employed for wheels for the duration. Phenolic resin wheels prove useful in the small sizes where quiet operation is desired for manually-propelled equipment. Other special types include canvas covered wheels to prevent marring, floating hubs, grooved rims (*Fig.* 5), etc.

Hub designs for axle mounting may be grouped into six typical types as indicated in *Fig.* 7. The hub for a plain axle is the simplest and cheapest mounting. It is utilized for equipment where ease of movement is not a major feature and where refinements are not warranted. The taper axle is essentially the same, under operating conditions, as the plain axle except that it may be adjusted to a closer running fit and may also be adjusted for wear. Antifriction bearing designs included in *Fig.* 7 are ball, needle, roller and tapered. The roller bearings are capable of transmitting high radial loads at moderate speeds. Tapered roller bearings are preferred for service where side thrust is also encountered.

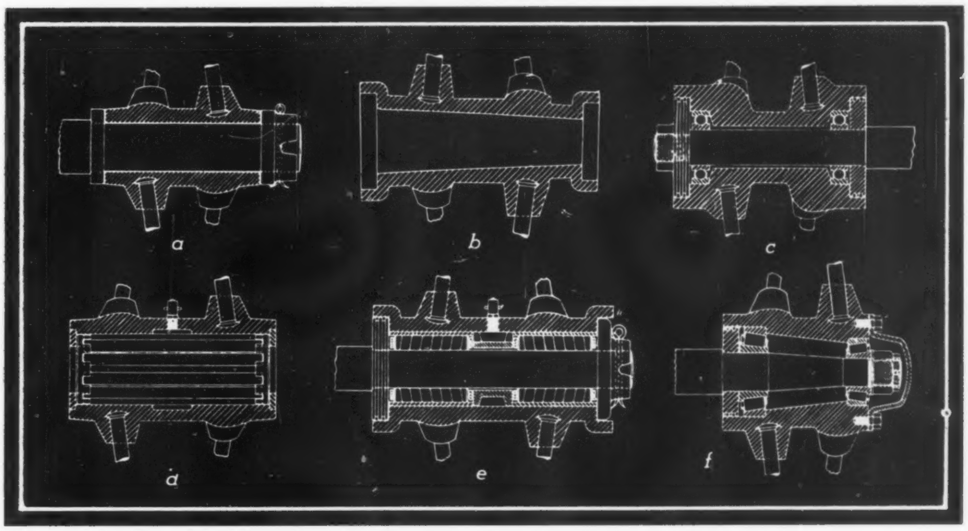


Fig. 7—Types of hubs. At (a) is plain-end hub for plain axle, (b) taper-axle hub, (c) ball bearing hub, (d) needle bearing hub, (e) hub for spiral roller bearings, (f) hub for tapered roller bearings

Referring to *Fig.* 5, grooved wheels have been the answer to many problems in aircraft assembly lines where wheel tracks formerly were channels in which the wheels rolled. Debris, chips and tools often would jam in the tracks or the wheels would bind against the sides of the channels. With the V-grooved wheel, inverted angle irons now serve as track, obviating the trouble formerly experienced. Cotter pins, chips, etc., fall off the track, allowing free rolling.

Grooved wheels are also supplied as standard in both swivel and nonswivel types of casters. Part II of this series will discuss selection of these and other casters, their mountings and arrangements for various conditions of loading and maneuverability.

Helpful cooperation of the Geneva Metal Wheel Co. in supplying information included in this article, as well as *Figs.* 2 and 7, is particularly acknowledged, as is the cooperation of The Colson Corp., *Fig.* 6, French & Hecht, Inc., and Picker X-Ray Corp., *Fig.* 1.

Vibration Isolation i

By Paul C. Roche
Field Engineer
Lord Manufacturing Co.

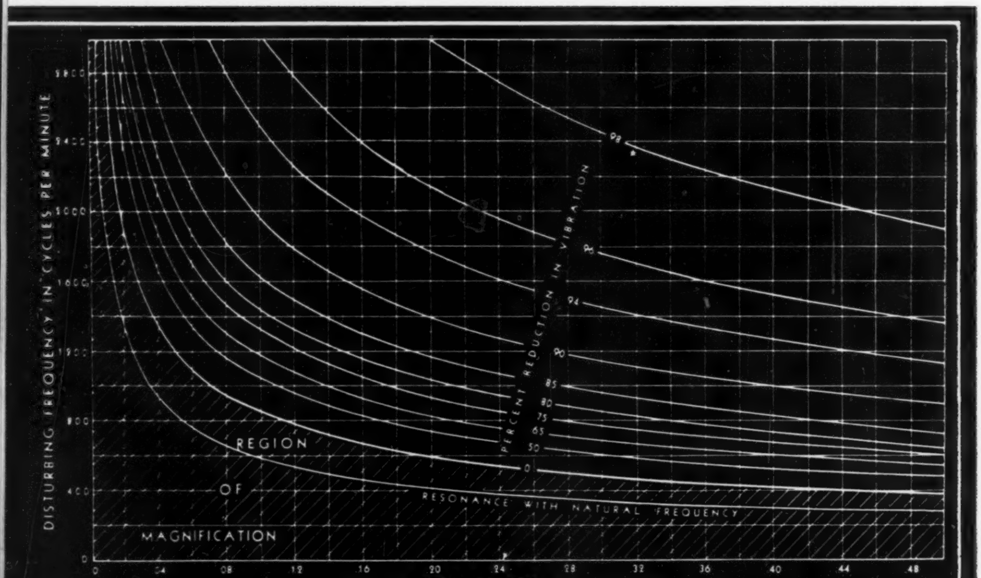


Fig. 1—Mounting efficiency in terms of disturbing frequency and static deflection

PROTECTIVE mountings applied to military or industrial equipment fall within either of two categories: (1) Those used to isolate, from the supporting structure and surroundings, disturbances originating within the mounted equipment; and (2) those used to support delicate equipment, protecting it from disturbances.

Typical of the first case are rubber suspensions used between the engine and plane structure on modern military and commercial aircraft. Illustrative of the second case would be an instrument panel, radio, or bomb-sight resiliently mounted within that same airplane. Inasmuch as the first case has been comprehensively treated in recent technical papers, the present discussion will concentrate on the second case. It so happens that quantitative analysis follows the same fundamentals in both cases, although many of the factors are individual to each case.

The types of disturbance from which protection must be provided are commonly known as steady state vibration and shock. Common sources of vibration are main propulsion systems, auxiliary power plants, and rotating or reciprocating machinery, the vibration characteristics varying with speed of operation and condition of loading. Such oscillatory vibration also occurs in an Army tank lumbering along at steady speed due to the recurring contact of individual treads with the suspension bogey wheels and with the ground. Wheel-driven vehicles may experience similar regular vibration from contact with seams in concrete pavement or the peaks of "washboard" terrain.

Maneuvers of military craft over irregular ground can also excite aggravating shocks. Moreover, impacts encountered in military craft can be of ballistic origin, due to discharging guns mounted on the craft or to enemy shells. Torpedo hits and near misses, as well as mine explosions, are responsible for particularly violent shocks. The isolation of forces which would destroy delicate equipment does not constitute the sole aim in adopting protective mountings, as the equipment itself is often rugged enough to take it, from the strength standpoint. Sensitivity of operation has to be preserved, particularly in the case of relays, circuit breakers, and voltage regulators, the contacts of which may open or close at inconvenient times due to disturbing influences.

Whereas oscillation at natural frequency for a short period of time is characteristic of the effect of shock, excitation of that same system by a source of steady state vibration will cause oscillation of the mounted system at the rate of the impressed frequency. Because the selection of most desirable mounts to isolate

Fig. 2 — Right — Aircraft instrument panel using two mountings in series. Vibration of structure on which panel is mounted is evident from blurred photographic image

*Abstract of a paper presented at the recent semiannual meeting of the A.S.M.E. in Los Angeles.

War Machines*

a predominant condition of *vibration* is often based on procedure entirely different than for *shock* protection, a logical question arises as to where the line is to be drawn between the two conditions of disturbance as frequency of impulses becomes lower and lower. The deciding factor is whether or not the mounted equipment comes fully to rest between successive impulses; if it does, the condition may be treated as shock; if not, vibration prevails.

Concentrating for the moment on vibrations such as may occur in an aircraft instrument panel, it is to be recognized that these may be of a translational or rotational character. The resiliently mounted system itself has six degrees of freedom—three translational and three rotational. When the word "freedom" is used in this sense it should be remembered that it is merely relative and that movement might be brought about with less effort in one direction than in another, depending upon the characteristics of individual installations.

If a body were free-floating in space it is readily understandable that the magnitude of vibratory force reaching it from any outside source would be nil. Of course it is impossible to realize this ideal condition with a body mounted to a structure but, by attaching through soft mountings, it is possible to reduce transmitted vibratory effects considerably. Naturally a resilient mounting will sustain a steady deflection due to the weight of the equipment supported; then, as vibratory impulses from the

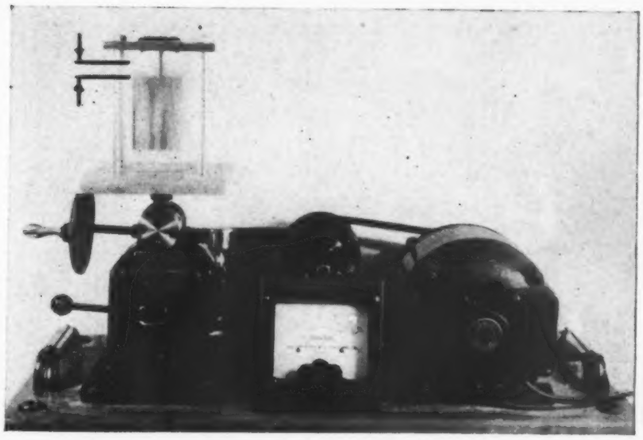


Fig. 3—Positive displacement exciter is used for tests on rubber-mounted bodies. Arrows indicate double amplitude of vibration at resonant frequency

structure are brought to bear, an oscillating motion occurs on both sides of this statically deflected position. The effect of such vibration is to accelerate the mounted equipment in the instantaneous direction of the impulse, with the comparatively soft resilient mountings developing restoring effort as a state of strain is set up. But, before the body has moved very far, the vibratory effort will have changed its direction; accordingly the only part of the disturbing effort transmitted to the mounted equipment is that involved in deflecting the resilient mountings a distance corresponding to the maximum relative movement between that system and the structure.

It would be folly to attempt an analysis of vibration control without first recognizing the types and frequencies of vibration impressed upon the particular system under consideration. Having mentioned that adequate protection of instruments in aircraft installations requires extreme care, this will be treated as a specific case. Lacking 100 per cent accurate balance of rotating parts, the engine may be expected to excite translational vibrations at one times crankshaft revolutions per minute (first order) in all directions in a vertical plane perpendicular to the crankshaft axis.

Also prominent will be a rotational trembling of the engine about the crankshaft axis as the individual combustion impulses tend to rock the engine intermittently in a direction opposite to crankshaft rotation. Occurring once every other revolution for each cylinder of a four-cycle engine, this would correspond to 4½ order, or 4½ times crankshaft revolutions per minute for a 9-cylinder engine. Analysis of the firing cycle will reveal a component occurring at crankshaft speed, which accounts for a first-order rotational vibration effect. Any misfiring cylinder in such an engine will create a power irregularity every other revolution, in which case a half-order rotational vibration would be apparent. These several modes and frequencies of disturbance are known to occur simultaneously and continuously.

With commonly encountered aircraft engines the idling speeds may occasionally be as low as 400 and up to 800 revolutions per minute. Cruising and top speeds cover the range 1200 to 3000 revolutions per minute, the operating speeds for liquid-cooled engines extending into the higher part of the speed ranges. Inasmuch as the translational vibrations are inertia effects with forces varying as the

square of the speed while the rotational vibrations are not of great magnitude when the engine is not delivering power to the propeller, it has not been found necessary to give great concern to vibrations at frequencies corresponding to low idling speeds.

So much for impressed frequencies of vibration. An expression for the natural frequency of a resilient mounting system is:

$$F_n = \frac{188}{\sqrt{D}} \qquad (1)$$

where D = static deflection, inches, existing in the mountings under the equipment weight supported, and F_n = natural frequency of the system, cycles per minute, in vertical translation. This formula can be applied to compute horizontal translational natural frequency as well by substituting in it an apparent deflection which is to be visualized as the amount rubber would deflect if actually

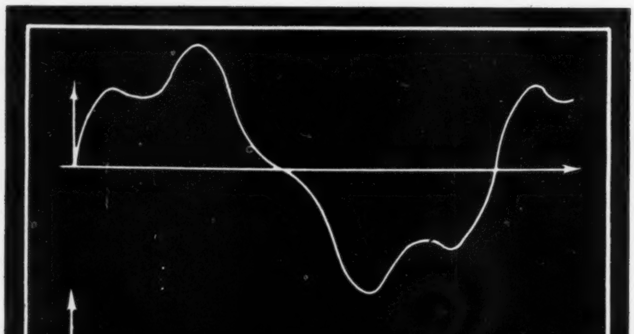


Fig. 4—High frequency wave (middle) superposed on low frequency (bottom) results in combination wave (top) typical of actual vibration conditions

supporting the same equipment weight on a unit of stiffness corresponding to that prevailing in the horizontal direction.

The relationship between natural frequency of vibration and the frequency of disturbing vibration from the engine occurs in the expression:

$$T = \frac{1}{\left(\frac{\text{disturbing frequency}}{\text{natural frequency}}\right)^2 - 1} \qquad (2)$$

where T is the transmissibility factor or that portion of the amplitude of vibration that is transmitted through the resilient mountings to the supported equipment. It should be noted that solutions of T less than, equal to, or greater than unity may result, but only when it yields a decimal value does vibration reduction occur. *Fig.* 1 is based on this formula in which the degree of vibration reduction may be observed by taking into account frequency of disturbing vibrations from the engine, and static deflection in the mountings under the equipment weight supported. This clearly illustrates that, by an intelligent choice of static deflection rating in line with the disturbing frequencies encountered, the possible degree of vibration isolation is tremendous. It may also be seen that, under certain poor combinations of conditions, the presence of the mountings may fail to reduce vibration transmission, and may even aggravate or increase it.

This latter case is known as resonance or sympathetic vibration, and when it occurs the natural frequency of the mounted system responds in step with the disturbing frequency of vibration from the engine, theoretically causing the amplitude of vibratory motion to become infinite. Particular care should be taken to insure that resonant speeds, of which every resiliently mounted system has one for each degree of freedom, occur only when excited at an engine speed below the operating range. If this is not entirely practical, it should at least be made to occur at a low idling speed or a speed known to be transient, not a speed at which the engine may operate continuously, such as take-off, cruising, or top speed.

Example Illustrates Use of Chart

One of six identical mounting points on the instrument panel of a current model two-engine Army bomber is shown in *Fig.* 2, providing a convenient opportunity for the application of the chart, *Fig.* 1. This illustrates vividly the effectiveness of an arrangement known as a double mounting, using one mounting over the other in series which results in double deflection. In this instance 1/16-inch static deflection per mounting gives ⅛-inch deflection per double mounting. With a disturbing frequency of 2400 cycles per minute, corresponding to top speed, it is found from the chart that about 95 per cent vibration isolation will result, so it is not surprising that the movement of the panel is imperceptible to the camera, despite the fact that the vibration in the supporting structure is of sufficient amplitude to show a blurred effect in that part of the picture.

It may also be deduced from the chart that vibrations at 1200 revolutions per minute, cruising, would be reduced by 75 per cent. Considering an idling condition at 600 revolutions per minute, while no reduction has been accomplished with respect to any low amplitude vibrations existing in this range, at the same time a resonant condition has been successfully avoided. Double-mounting installations like that shown in *Fig.* 2 constitute standard practice in the protection of instrument panels and other delicate apparatus in aircraft. A particularly desirable characteristic of this arrangement is that it can be so installed as to exhibit substantially equal stiffness with respect to horizontal and to vertical translation, by virtue of which the resonant frequency band is held to a minimum and isolation is insured for all three translational modes.

The question arises as to whether vertical translational

movement of a resiliently mounted body can be excited only by a vertical translational vibration from the source or whether rotational motion can occur only as the result of rotational disturbance. It has been observed that delicate equipment for aircraft service is actually so remote from the source of disturbance that the individual types of vibration tend to lose their identities, or become "coupled", as a result of which they do not reach the resiliently supported apparatus in their initial forms, translational disturbances predominating. Accordingly it is necessary to assume that any degree of freedom of the mounted equipment can be excited by any mode of vibration from the source.

It is possible to compute the performance characteristics of a resiliently mounted system (even to applying a correction for unsymmetrical mounting location) before making actual installation and to check such calculations by attaching the system to a test stand and subjecting it to a variable-speed source of vibration. *Fig.* 3 is a picture of such a test where the resonant frequency of a rubber-mounted body is in evidence. Oscillation of the table in this test device is imparted by a connecting rod driven by a rotating shaft through an adjustable double eccentric, providing means of varying amplitude as well as frequency of vibration. This is known as a positive displacement exciter. A reasonable double amplitude of oscillation for such a test would be 1/64-inch, comparing with .005 to .030-inch known to be conducted through aircraft structures. Amplitudes of 1/32-inch have been observed in airplanes in certain cases, which resulted in damage to structure as well as accessory equipment. Experience has shown that one can be certain of the ability of the most sensitive apparatus to stand up in aircraft service when its double amplitude of vibration does not exceed .004-inch.

In treating the subject of shock, close destructive ally of steady state vibration, it will be helpful to picture a structure which undergoes a sudden jolt or acceleration in response to an impact of external origin. As this occurs, the inclination on the part of a body resiliently mounted to that structure is to continue in a state of rest. If the mounting were so soft as to exercise no restraint whatever, and if sufficient clearance existed between that body and the structure, a large relative displacement could result and no part of the shock would be felt by the mounted equipment. A resilient medium, however, will build up a restoring force under strain, as a result of which it inevitably experiences a degree of motion. With certain reservations and proper consideration of all factors it may be stated that for outstanding shock protection, mountings should be chosen to be as soft as possible.

Mounting Must Provide Stability

This is consistent with the knowledge that the only force, and consequent acceleration, transmitted to the mounted equipment is that developed in the mountings as they deflect an amount equal to the relative displacement between the supported body and the structure. Before choosing an antishock mounting for minimum stiffness, however, it is necessary to provide proper static deflection to satisfy all conditions of steady state vibration that may exist at the same time. Moreover it is important to take care that undue instability of the mounted equipment relative to the supporting structure will not result, having allowed clearance to the extent of the intended unrestricted relative travel. It is a fact that in the selection of mountings to protect against shock, as well as in the control of regular vibration, extreme deflection beyond a reasonable degree fails to justify itself with appreciable additional protection.

The Tank-Automotive Center (Division of Army Ordnance Department), in cooperation with certain of its contractors, has carried on a large amount of shock test work. Shells have been fired at production models of tanks and at turrets, each containing delicate equipment mounted on representative contrasting types of cushions.

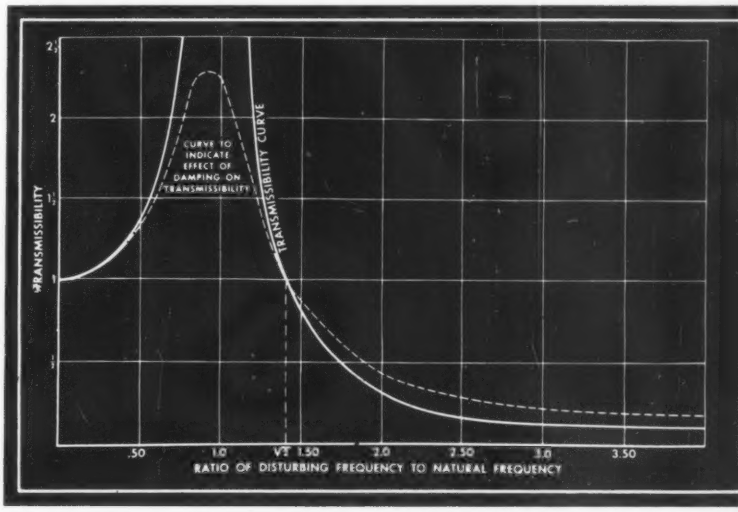


Fig. 5—Damping increases transmitted vibration in working range of application of resilient mountings, reduces it in amplification zone

The shells used correspond to the most severe types fired against our tanks in warfare and provision was made to insure that the projectiles would not penetrate but would spend 100 per cent of their kinetic energy at the impacted structure. In most instances duplicate readings were taken of displacements, frequencies, and amplitudes of movements in the tank armor and various other elements. After each round a careful visual check was made to record damage occurring.

Test records reveal that under impact the armor is subjected to high frequency motion of small amplitude, together with relatively low frequency motion of large amplitude. It is evident that the frequencies apparent are natural frequencies and harmonics of the tank structural sections which have been excited by the impact just as a tuning fork can be set in motion. However, resonance does not exist, since the excitation is not continuous but practically instantaneous. While the high frequency components are responsible for terrific accelerations that develop highly destructive forces in rigidly mounted apparatus, and thus create the principal need for protective mountings, the large amplitudes of the low frequency components are cause for concern since they may result in exceeding the safe deflection limits of the flexible elements. *Fig.* 4 shows diagrammatically the composition of the vibration excited, where high and low frequency components jointly constitute that shown at the top of the figure. Considering either component individually, x_0 expresses

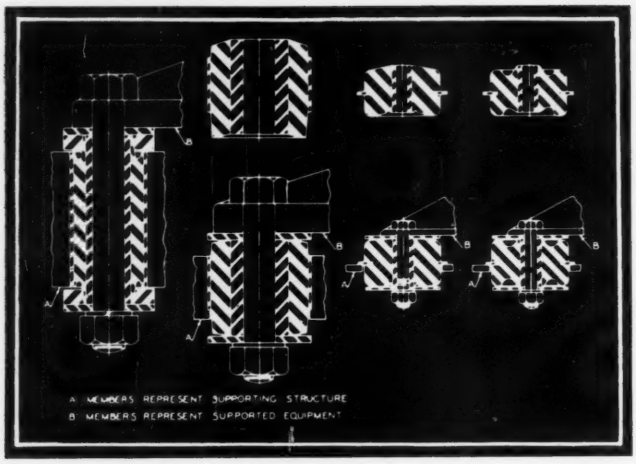


Fig. 6—Snubbing type mountings include conventional tube form (left), tube-form vertical snubbing (left center), sloping shoulder plate-form vertical snubbing (right center), and high shoulder plate-form vertical snubbing. Upper views show no-load shape, lower views normal loaded condition

the maximum displacement, feet, whereas x is the displacement after any time t, seconds; ω is the angular velocity of the motion—equal to cycles per second $\times$ 2π, giving radians per second. Such a sinusoidal relationship is analyzed according to expressions for simple harmonic motion as:

$$\text{Displacement} = x = x_0 \sin\omega t \quad \ldots\ldots (3)$$

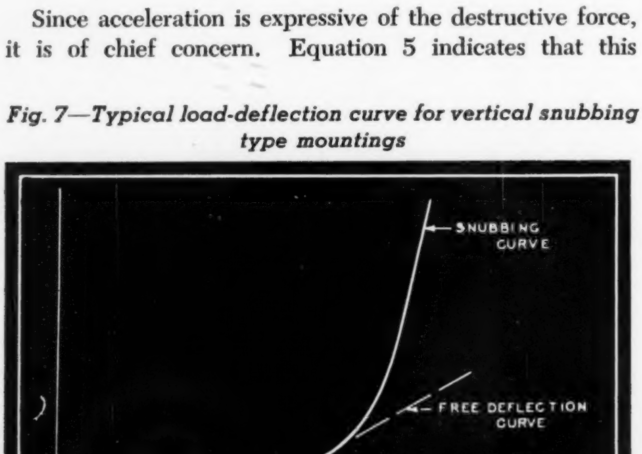
$$\text{Velocity} = \frac{dx}{dt} = x_0\omega\cos\omega t \quad \ldots\ldots (4)$$

$$\text{Acceleration} = \frac{d^2x}{dt^2} = -x_0\omega^2\sin\omega t \quad \ldots\ldots (5)$$

Since acceleration is expressive of the destructive force, it is of chief concern. Equation 5 indicates that this reaches a maximum value equal to $-x_0\,\omega^2$. Solutions of this expression are commonly divided by 32.2 feet per second per second, enabling reference to accelerations as some multiple times g, that is, times acceleration of gravity. High values of acceleration, as great as several thousand g, may exist in the structure, but these are not of great significance when considering forces affecting resiliently mounted equipment. It must be recognized that no accelerations anywhere near the peak figures prevail over a sufficiently long time interval to permit that amount of relative motion to occur which would develop correspondingly high restoring forces in the flexible elements. Actually the structure will have changed its direction of motion before very great relative motion can take place.

In terms of Equation 2 the ratio between disturbing frequency (from the structure) and natural frequency (of the resilient mounting system) is sufficiently high to give a low transmissibility factor when considering the high frequency components, while some higher, though not necessarily unreasonable, transmissibility will result when substituting the lower frequency components of structural vibration. The benefits of a resiliently mounted installation over a rigid system in the presence of these influences is obvious.

Fig. 7—Typical load-deflection curve for vertical snubbing type mountings



Mounting Stiffness Controls Transmitted Force

Typical of the higher frequency oscillations just mentioned was that set up in the hull of a light tank during ballistic impact testing, when a 9000-cycle per minute disturbance was recorded under the fire of a gun similar to that used as the principal weapon on American light tanks. Accelerations caused by terrain shocks and gun recoil do not exceed 4g, usually falling in the range of 2½ to 3½g. In a case where definite information is available as to the degree of acceleration that a piece of equipment can withstand without damage or faulty operation (which it is believed should be at least 10g if ballistic impact is likely to be encountered), and where records indicate the magnitude of vibratory forces impressed on that body by the structure, there is wisdom in adopting mountings of such stiffness that the force transmitted to the mounted system is just safely incapable of causing damage or unsatisfactory operation. Such a choice would be made in an instance where unnecessary softness had to be avoided for reasons of stability, etc. In arriving at such a compromise it should be borne in mind, however, that resonance produces much more severe accelerations at high frequency than at low frequency, should any possibility of resonance exist.

Use Shock Isolation Capacity of Structure

It has been agreed, on the basis of these tests on tanks, that instruments are incapable of measuring, and it is impossible to picture with any hope of accuracy, what happens at the point of ballistic impact or within a radius of at least a foot. To go a step further with this thought, it is known that the impact intensity of transmitted shock is reduced upon travelling through the armor, which is the basis of a recommendation to avoid attachment of equipment at positions on the tank that may be exposed to direct shocks by mines, bombs, or shell fire. The tank

structure has an inherent capacity for shock isolation, of which advantage should be taken. By the same token, greater benefit can result from mounting equipment on a comparatively flexible component of the tank such as the turret basket. When it is impractical to avoid mounting equipment to the hull there is said to be an advantage incident to the use of brackets allowing a degree of permanent deformation under close-by impacts, which permits relative motion without increasing the force being transmitted; no part of the mounted equipment should be closer than one inch to the hull, since displacements of this magnitude can and have occurred under ballistic impact.

As in the other services the Navy's problems are unique. To compete favorably in technical warfare involves the adaptation of a large amount of specialized apparatus. With installation space at a premium, due to limited clearance in many ship locations, the Navy is constantly confronted with a problem entailing restriction of the overall travel of certain antishock mounted equipment. Consistent with a plan of providing protection from the most aggravating shock components encountered in naval vessels without permitting unreasonable equipment travel, it is becoming common practice to select rubber mountings for static deflections of .01-inch or less, corresponding to resonance at a frequency above that impressed by the main propulsion system of the ship, which seldom exceeds 1600 cycles per minute.

Isolation and Absorption Defined

Two methods of reducing vibration transmission are those referred to as "isolation" and "absorption", and the difference between the two expedients is clear cut. Isolation implies the employment of resilient cushions which store vibrational energy during one part of a cycle and release it during a later phase; all of the foregoing treatment has been based on this principle. On the other hand a mounting is said to absorb vibration when it receives kinetic energy during all parts of the vibratory cycle yet releases none mechanically, converting it all to heat. Very often both these means are incorporated in the same system to accomplish vibration reduction jointly, as a shock absorber supplementing the spring of a vehicle. Absorption is described as having a damping effect on a vibrating system and is capable of reducing the motion of the mounted system when the action of resilient elements would increase it, as in the case of resonance, transient effects, or rebound. At the same time, whenever the resilient mountings reduce the vibration, the damping effect detracts from the effectiveness of such reduction.

A mild degree of vibration absorbing capacity exists in rubber, on the order of 4 per cent of that which would be required to eliminate the vibration in one-half a vibratory cycle, that is, 4 per cent of "critical damping". *Fig.* 5 shows the influence of damping on transmissibility. It is important to note that its effect is favorable at frequency ratios of less than 1.414, where more pronounced vibration magnification would otherwise take place, and unfavorable above that value, where more effective vibration reduction would otherwise result. The limited absorptive capacity in rubber is akin to fluid friction, the effect of which increases with the velocity of excitation, exercising negligible frictional restraint at the beginning of motion. The addition of absorption to a resilient support can, under certain circumstances, prove to be of benefit, but attention should always be directed to insure that all advantage has been taken of mounting stiffness best suited to existing conditions before increasing the damping effect.

Frictional damping is not the only means employed in controlling large displacements due to impacts or resonant vibration in a spring system. Another device in common use is that known as a snubber which has been built into mountings integrally and is referred to in this case as a vertical snubbing design. It becomes a justifiable matter of habit, in the course of working up mounting designs during peacetime, to adopt a vertical snubbing mounting for, say, a power plant in an automotive vehicle and to conclude that shock control had been provided. Actually this is an entirely different case from those discussed in this article since the purpose in such nonmilitary installations is to limit overall movement upon encountering road shocks with the sole aim of preserving alignment between

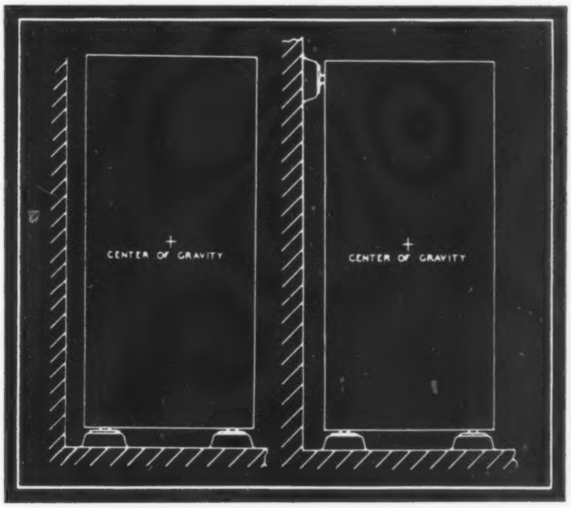


Fig. 8—Undesirable (left) and desirable (right) methods of mounting tall equipment. Latter provides necessary stability

driving and driven equipment, and between power plant and frame. Reduction of forces due to such impacts was not demanded of the engine mounts because chassis springs, operating at far greater static deflection, performed this function effectively.

Various commercial forms of snubbing type mountings are shown in *Fig.* 6, all of which are proportioned to provide a small amount of travel for accomplishing vibration isolation within a predetermined working range and to limit overall movement beyond that range. *Fig.* 7 is a typical load-deflection curve for this type of mounting and it is to be realized that such a unit usually appears in a shape that is offset toward one end when not supporting load, in order that the application of its intended static load will bring it into symmetry with equal clearances adjacent to restricting surfaces at both "load" and "rebound" ends. Because the stiffness of the mounting is so much greater outside the working range, it is imperative that a snubbing type mount be selected only on the basis of ac-

(*Continued on Page* 204)

Controlling Properties of Powd ron

By F. V. Lenel
Moraine Products Div., General Motors Corp.

POWDER-METALLURGY products may be divided into two classes: The first includes those products which cannot be made by other methods or at least cannot be made easily. In this class belong refractory-metal wire and sheet, cemented-carbide tools, self-lubricating bearings, electrical-contact materials, etc. The second class consists of those parts which can be made by conventional methods such as die casting or machining of wrought or cast metal, as well as by powder-metallurgy methods. It is this latter class with which this article is concerned.

It is often possible to briquet and sinter to the finished size. In this way machining and scrap are eliminated. To make the powder metallurgy competitive, the savings effected must be large enough to compensate for the increased cost of raw material, and the higher cost of the powder-metallurgy processing operations, namely, briquetting, sintering and sizing.

In some cases, where only a semifinished product can be briquetted, the cost of the final machining will have to be added to the powder-metallurgy operations in order to arrive at comparative costs. A complete comparison of costs, based upon the number of pieces to be produced, should also include the cost of tooling for briquetting and sizing, as against the tooling setup for machining by conventional methods. There are usually fewer operations in the powder-metallurgy process, but the cost of the hardened-steel tools needed for these operations is not low. Furthermore, of course, it is necessary that the parts have mechanical properties which will be satisfactory for the service for which they are intended.

Reference will be made particularly to parts which are manufactured from iron powder and which, therefore, compete with those machined from steel bar stock, steel forgings, or steel or iron castings. In the past many claims have been advanced with regard to the mechanical properties of such iron parts, and it has been felt throughout the industry that a clarification is necessary as to what mechanical properties can be obtained consistently in large-scale production.

Formulating Powder Metal Specifications

Because of the widespread demand for specifications for powder-iron parts, the manufacturers of metal-powder products have organized a committee to prepare general specifications. The formulation of these specifications has

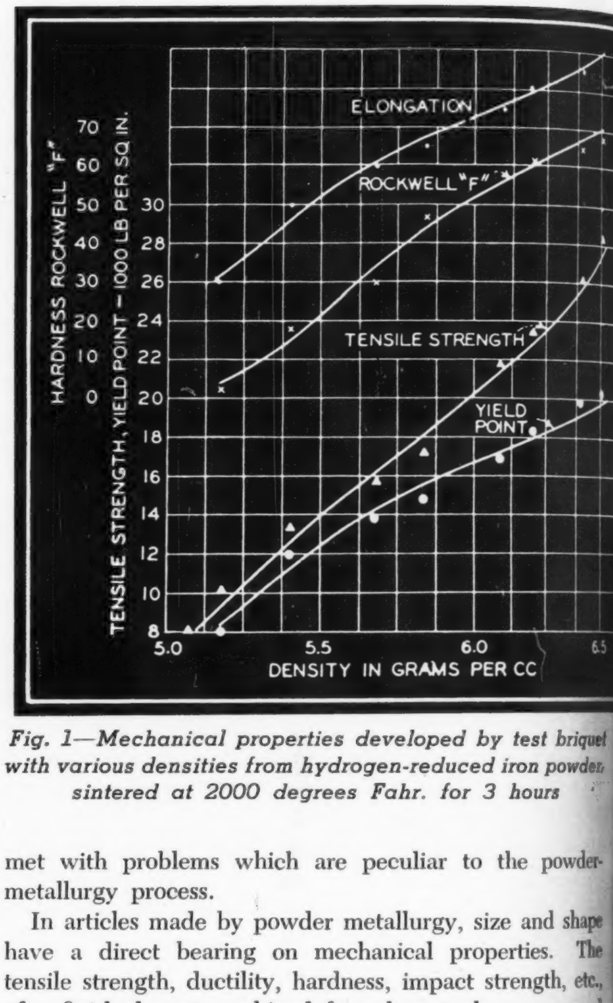


Fig. 1—Mechanical properties developed by test briquet with various densities from hydrogen-reduced iron powder, sintered at 2000 degrees Fahr. for 3 hours

met with problems which are peculiar to the powder-metallurgy process.

In articles made by powder metallurgy, size and shape have a direct bearing on mechanical properties. The tensile strength, ductility, hardness, impact strength, etc., of a finished part machined from bar stock are assumed to be the same as those of the stock from which the piece was made. In castings, coupons can be cast together with the casting, the mechanical properties of which can be determined, although the influence of section thicknesses may be important and will have to be taken into consideration. In powder-metallurgy, effects of size and shape upon mechanical properties is even more accentuated and will be discussed in detail to show the reasons.

Metal-powder parts are briquetted in steel dies and sintered in furnaces with controlled atmospheres. After sintering, further processing may involve a sizing operation or sizing and subsequent resintering. For purposes of this discussion, the mechanical properties of the finished pieces may be described as depending upon four factors, as follows:

1. Composition of the powder mixture
2. Grade of powders used
3. Temperature, length of time and atmosphere of the

*From a recent paper presented before the American Society of Mechanical Engineers.

ron Parts*

sintering and resintering operations

4. Density, resulting from briquetting, sintering and sizing operations.

The first three of these factors have parallels in cast and wrought materials (chemical composition, purity, heat treatment) but the fourth is peculiar to metal-powder parts. Density is also different from the other factors, because it depends, not only upon the processing, but also upon size and shape. This correlation between density, processing, size and shape and mechanical properties may be treated under the following headings:

1. How mechanical properties depend upon density
2. How density depends upon processing steps
3. How density depends upon size and shape
4. How size and shape influence processing.

Tensile strength, impact strength and hardness, as well as the ductility, of a powder-metal part will increase with increasing density, if the same powder mixture, quality of

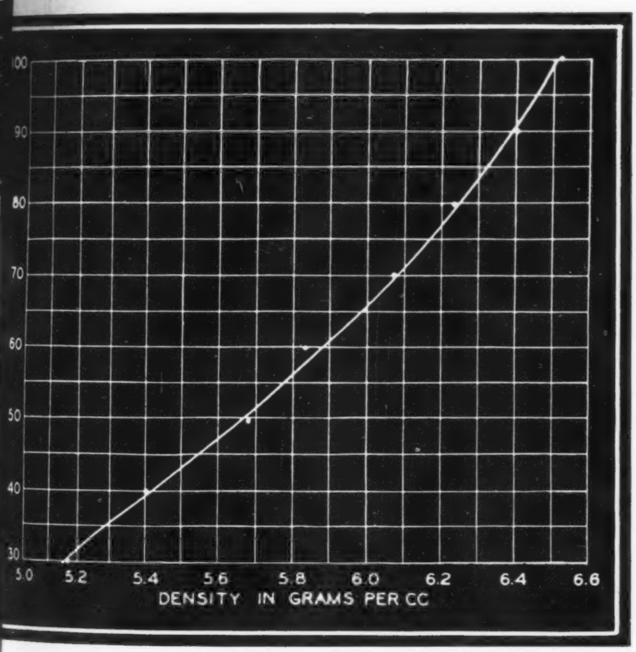


Fig. 2—Effect of briquetting pressures on density for test pieces in Fig. 1

powder and sintering treatment are used. *Fig.* 1, giving the tensile strength, yield point, elongation in one inch, and hardness for a number of test specimens made from a grade of hydrogen-reduced iron powder and sintered three hours in partially combusted natural gas at a temperature of 2000 degrees Fahr., but having different densities, illustrates the point.

Density of a powder-metal product will depend to a certain extent upon the powder mixture, the quality of the powder, and the heat treatment of the product, but the most important factors influencing the density will be the briquetting pressure used to compact the powder into a coherent piece, and the sizing pressure used in coining or cold-forging after sintering. *Fig.* 2 shows the correlation between briquetting pressure and density for the same set of specimens as shown in *Fig.* 1.

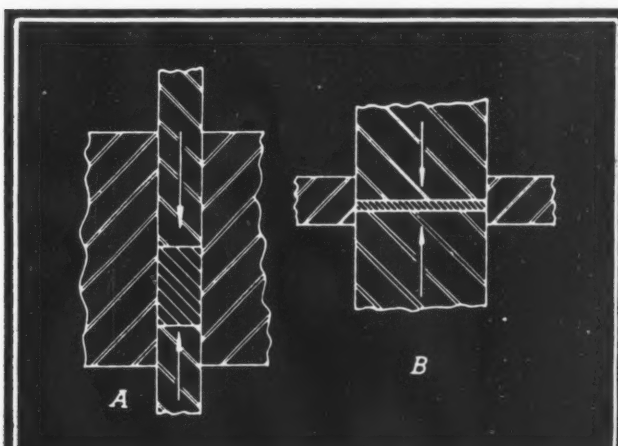
The density of pieces having different shapes and dimensions will be different even if the same briquetting and sizing pressure is used. These differences are due to the fact that most powder-metal parts are briquetted only along one axis. Therefore, the friction between the die and the powder will influence the density in two ways: With a given briquetting and sizing pressure, pieces which are long and slender in the direction of pressing will be less dense than pieces which are short and thick, *Fig.* 3. Pieces whose ratio of length (dimension in the direction of pressing) to width (dimension perpendicular to direction of pressing) is large (in general greater than two) will have an uneven density over their length. If pressed from both ends they will be denser at the ends than in the middle, and if pressed from only one end, they will be denser at one end than at the other, *Fig.* 4.

Nonuniform Sections Require Multiple Punches

Pieces which have a nonuniform section in the direction of pressing (e.g., flanged pieces) may either be made with single punches or with multiple punches sliding relative to each other. If the piece is made with single punches the density will be higher in the short section than in the long section, because of the difference in the rate of compression of the powder. For some applications this difference in density and the corresponding difference in strength, ductility and hardness is permissible; for other pieces multiple punches are necessary to make the density throughout the section as even as possible, *Fig.* 5.

As explained, a high density may be obtained by using high briquetting and sizing pressures. However, these pressures are limited by the pressure which the briquetting and sizing dies will stand without rapid failure by fatigue or wear. The permissible sizing pressure and, to a lesser extent, the permissible briquetting pressure will also de-

Fig. 3—Lower density of long slender part A compared with short, thick part B



pend upon the shape of the piece and the design of the die. A punch having a thin weblike section will not stand as high a pressure in pounds per square inch as a solid heavy section. For large pieces, the capacities of the available briquetting and sizing equipment may also be limiting factors. The density which can be obtained in complicated pieces will, therefore, not be as high as it is in standard test pieces. In some cases, the lower physical properties, due to this lower density, may be compensated for by a change in the grade of powder or the sintering conditions, but such changes will, of course, also alter the cost.

The foregoing discussion shows why it is not practical to set up specifications for tensile strength, ductility, etc.,

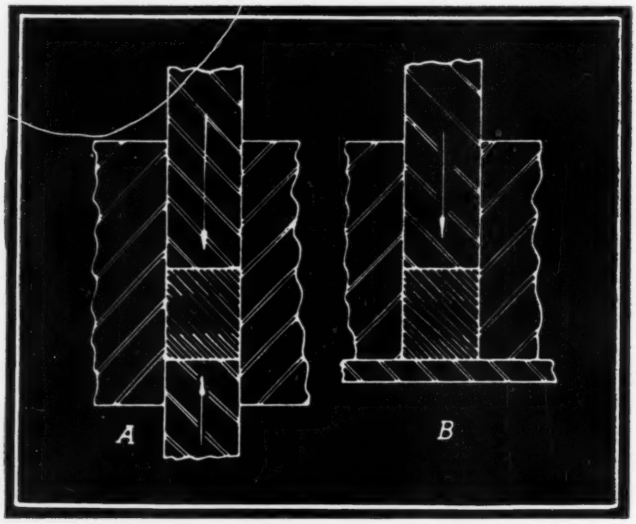
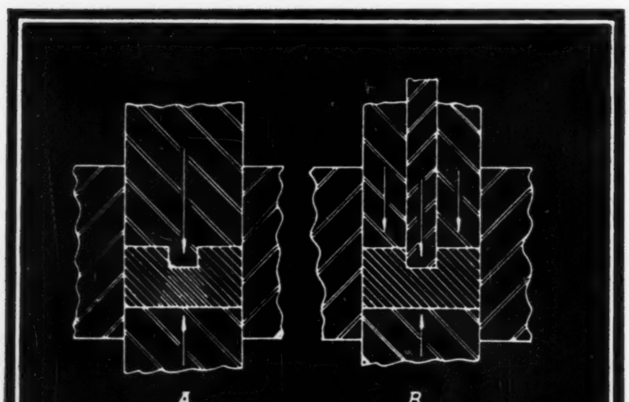


Fig. 4—Above—Uneven density distribution for parts briquetted from both ends at A and from one end at B

Fig. 5—Below—Uneven density distribution in part with nonuniform section when briquetted with single punch at A, compared with even density of similar part briquetted with multiple punches



to be determined on specially briquetted test specimens. The properties of such test specimens would not be representative of the properties of a complicated briquetted piece, even if the same briquetting pressure and sintering conditions were used. Neither is it usually practical to cut tensile or impact specimens from the metal products themselves, because most parts made from powders are too small for cutting test specimens. For this reason it does not seem advisable to set up general specifications for the tensile properties of powder-metal parts. The only specifications for iron parts in actual figures which should be included in a general specification are those for chemical composition and density. In order to grade parts made from iron powder, according to their properties, they may be divided into three types, as follows:

TYPE A: Materials having mechanical properties similar to common cast iron, suitable for applications where the stresses are low

TYPE B: Materials similar to TYPE A, having improved tensile strength, a definite yield point and a noticeable elongation

TYPE C: Materials having mechanical properties approaching ordinary malleable iron, suitable for applications where stresses including impact are moderate.

There would be no sharp lines of demarcation between these various types, and the division into the three types would be for the convenience of the user rather than for any fundamental reasons. The grade of powder used, the briquetting and the sizing pressures, the time and temperature of heat treatment would determine in what class a particular product would fall, changes in one or more of these factors possibly shifting a product from one class to another. The three types would be identified by their specifications for chemical composition and for density, as suggested in TABLE I. The table shows that for parts with better physical properties, namely, higher strength and ductility, an impure product with low density would not be suitable. The purity rises from 95 per cent minimum iron to 98 per cent minimum iron; the density from 5.4 to 6.5 grams per centimeter minimum.

TABLE I

Chemical and Density Requirements

Type of Material	Total Carbon, (% Max.)	Total Iron (% Min.)	Density (g/cc Min.)
A	2.5	95.0	5.4
B	0.4	97.5	5.8
C	0.2	98.5	6.5

Type A would include parts which contain up to 2½ per cent carbon. The carbon may be added to the powder mixture in the form of graphite and would combine with the iron during the sintering operation. Such carbon-containing parts are harder and stronger than straight iron products but are also more brittle.

Strength Tests for Individual Parts

Because a general specification necessarily has to be broad, it will be necessary to develop detail specifications for individual parts. They should be set up after performance tests on the parts have shown that they are satisfactory for the application. The detail specifications should furnish a check on uniformity of production and, whenever possible, on performance characteristics. Such tests are as follows:

1. Crushing, radial or shear strength
2. Bending or impact drop test
3. Hardness at one or more specified points of the piece or hardness variation over the entire piece.

Static strength tests would be suitable for pieces where no particular impact resistance is needed, for instance, oil-pump gears may be so tested. The gear is held in a fixture and the load required to break off one of the teeth is determined. Hollow cylindrical pieces may be tested by a radial break test; the piece is crushed between two flat plates and the load is measured when failure occurs.

When a certain amount of shock resistance is required in a piece, a bending or an impact test should be devised. Tests have shown that impact strength and ductility parallel each other. In other words, a material which does not show any elongation in a tensile test, or no bending in a bend test will usually be shock sensitive. On the other hand, a material which has a fair amount of ductility will not break so easily under a sudden blow.

In *Fig.* 6 is shown a specimen in which a certain amount of shock resistance is needed. This piece is briquetted with the long section in the direction of pressing, pressure being applied both from the top and the bottom. As previously explained, this piece will show some difference in density along the long section. It will have the lowest density and, therefore, the lowest strength and ductility in the middle of the long section. The test fixture is so designed that the piece is bent to a specified degree at this, its weakest point, the idea being that if the piece does not fracture at this point it will be satisfactory throughout. Simultaneously, the point where the flangelike section and the long section join is tested by bending the flangelike section. As previously outlined, the bend test will also give a good indication of the resistance to shock of the piece.

Hardness Tests Need Interpretation

Indentation hardness tests, like brinell or rockwell, are convenient for powder-metallurgy products, but it is necessary to understand the mechanism of an indentation test on porous material. The resistance to indentation is a function not only of the chemical composition and the microstructure of the tested piece but also of its density. The lower the density the further will the indenter sink into the material and the lower will be the reading. Two pieces which show identical hardness readings may, therefore, have entirely different properties. One may be low in combined carbon but high in density and be therefore ductile and shock resistant. The other may have a high combined carbon content but a low density and would, therefore, be quite brittle and shock sensitive.

The influence of density on the indentation-hardness reading must also be considered when the abrasion resistance of a material is to be evaluated. Abrasion resistance depends much more upon the microstructure of the material than upon its density. For instance, an iron-powder piece containing sufficient combined carbon may be heat treated and may then be fully martensitic. Such a piece will be file hard, but the hardness reading on the rockwell C scale may be only 30. The peculiarities of the indentation-hardness test should always be kept in mind, particularly so when the hardness of a powder-metal part is compared with that of one made in the conventional way.

TABLE II

Tensile Properties

Type of Material	Tensile Strength (psi)	Elongation (% in 1 in.)
A	15,000	½
B	25,000	3
C	35,000	7

Tensile Properties Not Included

For the reasons outlined, tensile properties should not be included as part of a general specification, however, such properties can be and have been measured. The values shown in TABLE 2 were determined on special subsize specimens. Typical values are given for the three previously suggested types of material.

It must be admitted that the values given in this table, which are purposely on the conservative side, are not very impressive. Under laboratory conditions, it has been possible to obtain much superior properties and some such values have been widely publicized, but at the present time no large-scale production technique has been developed which would permit the manufacture of parts from iron powder with high strength and high ductility at a competitive cost. The main difficulty lies in achieving satisfactory ductility and impact resistance, while it is much less difficult to obtain higher strength and high hardness by proper heat treatment. These heat-treated materials are brittle and cannot be used in any application where they have to withstand shock.

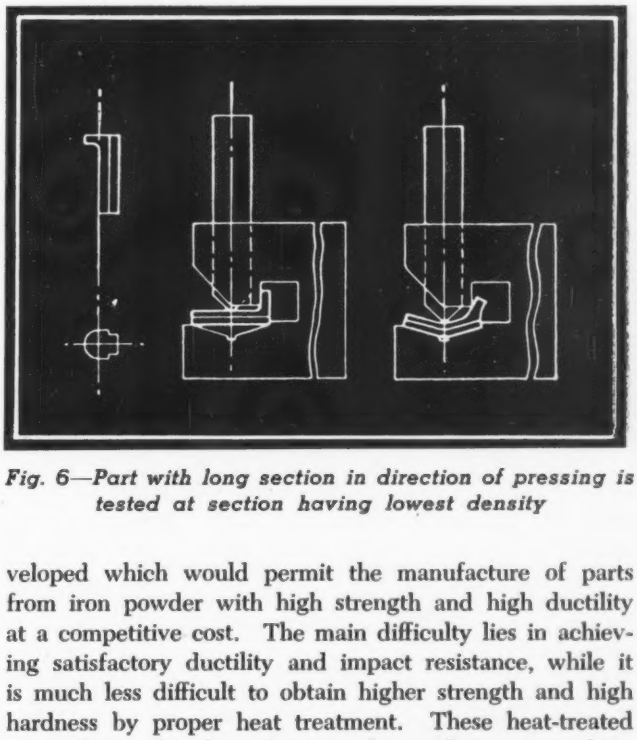
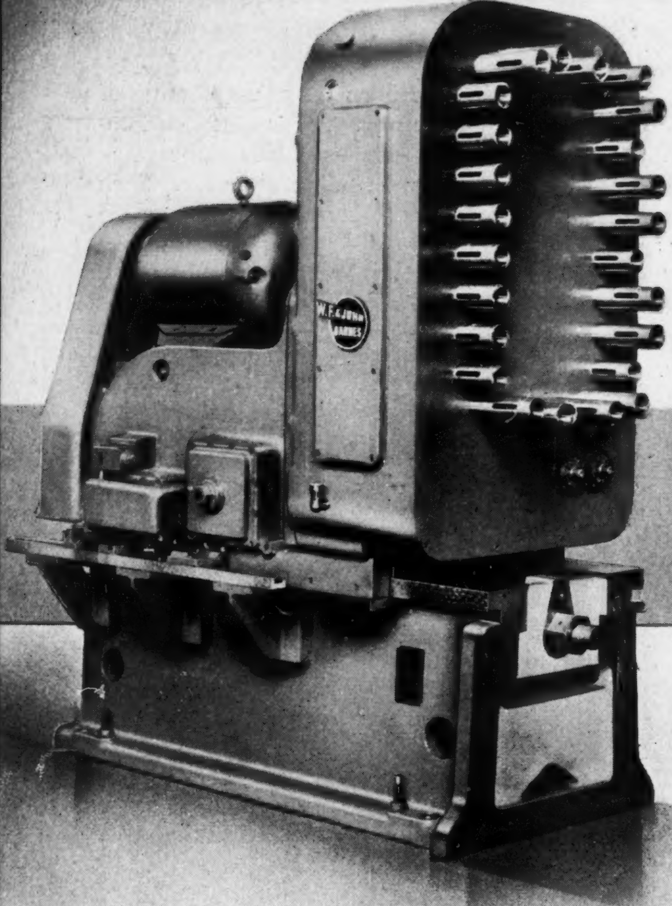
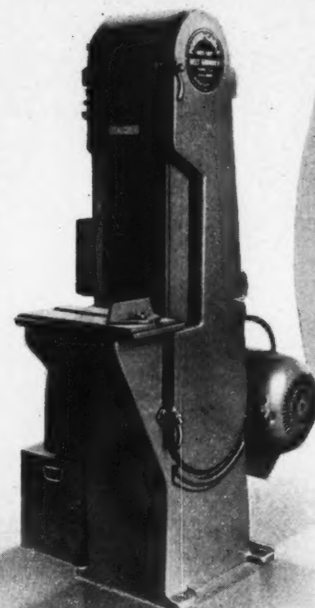


Fig. 6—Part with long section in direction of pressing is tested at section having lowest density

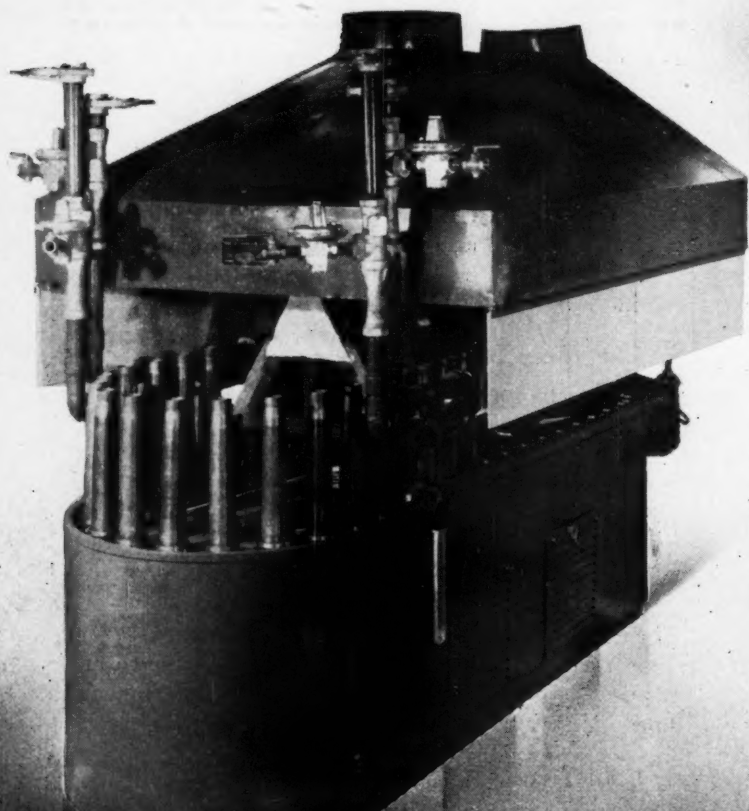
In many components of small arms or of expendable munitions, to mention only two fields which are at present in the foreground of interest, an accurate configuration is more important than high mechanical properties. Many of these components are at present low-carbon steel because that is the cheapest and most convenient material available. However, that does not mean that the physical properties actually needed are those of low-carbon steel. These are the applications where powder metallurgy is believed to have a field, if by this method a part can be made at a competitive cost and with a saving in machine time and in bar stock.



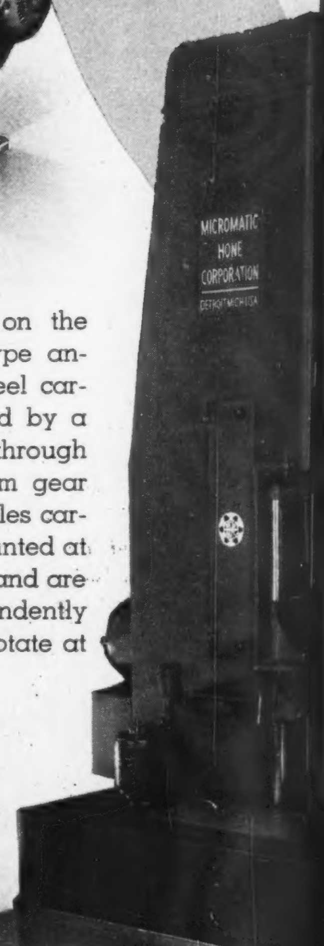
Left — Multiple-spindle head applied to Barnes standard hydraulic drilling unit is used in the drilling of holes for the revolving turret of the M-4 tank. Gears, spindles and shafts are heat-treated alloy steel

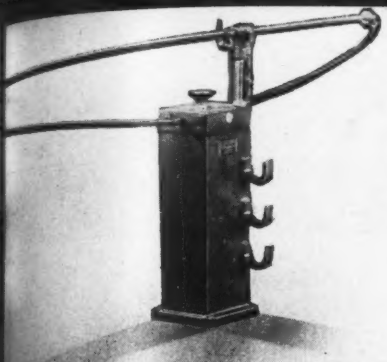


Right—Oscillating feed table with micrometer stop maintains accuracy and distributes belt wear on the Porter-Cable belt surfacer. For wet operations a controllable water system and double spray system are provided. Idler or upper pulley is mounted on a shaft supported on a dovetailed slide operated by a crank to adjust abrasive belt

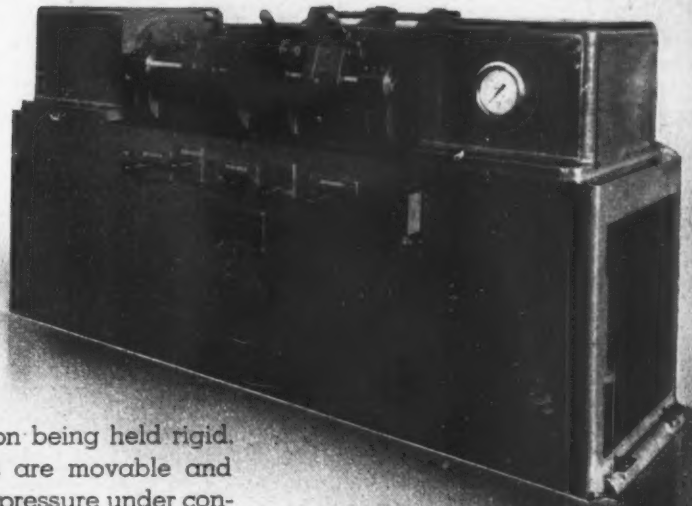


Left—Conveyor chain on the Morrison continuous type annealing machine for steel cartridge cases is actuated by a variable-speed drive through a totally enclosed worm gear reducer. Vertical spindles carrying the cases are mounted at each pitch of the chain and are rotated by an independently driven V-belt. Cases rotate at 60 to 100 revolutions per minute in front of the gas burners





...dle of the Gorton automatic ...hine is driven through flat end- ...g belt from a variable-speed ...unit permitting speeds from ...0,000 revolutions per minute. ...variable-speed camshaft and centrali... ...nel contribute to ease in ob- eration, ...cal interlocks prevent camshaft ...n spindle is not turning



Right—Airplane engine crankshafts are speedily assembled in the Denison horizontal crankshaft assembly press. Shaft is locked in the fixture in three sections, the center section being held rigid. The two end sections are movable and actuated by hydraulic pressure under control of the operator

Maces Behind Guns

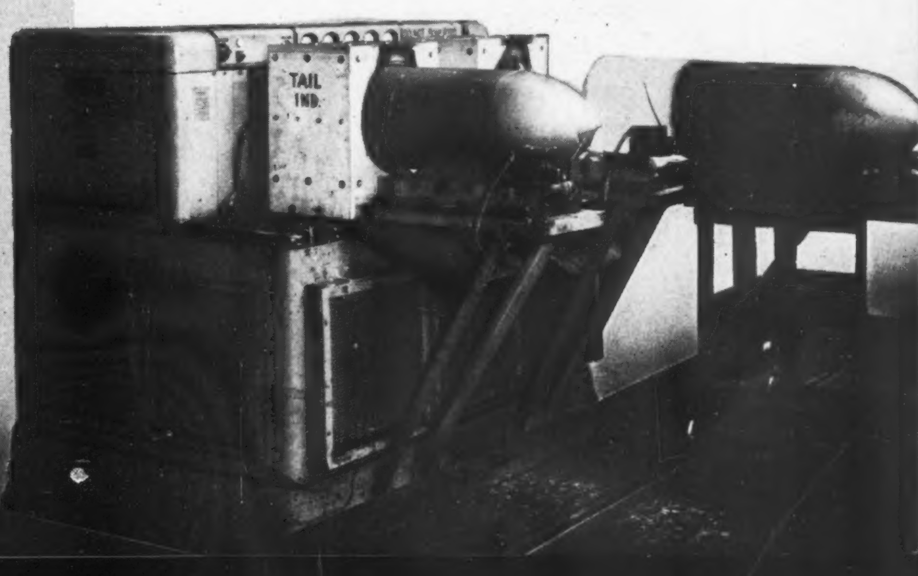
(...ze listings, see page 216)

Left—Loaded fifty-caliber cartridges are automatically inspected, chamber-checked and weighed by the new Sheffield Chekmatic. A plunger loads cartridges into chambers on an index wheel having dimensions identical with those on the weapons from which they are to be fired, rejecting those which will not fully enter the chamber under predetermined pressure

Left—Automatically giving correct voltage for the continuous range of currents available, the new Allis-Chalmers alternating-current welding machine simplifies operation at low currents because of high striking voltage, also saves power at high currents

Below—Combination of an automatically controlled spinner with Tocco induction heating enables one-piece 250 and 500-pound high explosive bomb casings to be shaped in only two operations—nosing and tailing. Because the heating machines could be placed close to the spinners it was possible to establish production line procedure

Left—Spindle of the Micromatic vertical honing machine is actuated by a belt-driven spline and also incorporates integral, mechanically actuated reciprocation of the tool. Hydraulic controls include automatic timer for cycle of spindle head movement, while electrical controls are entirely through pushbuttons



MACHINE *Editorial* DESIGN

Standard Parts Play Vital Role in War Machines

FEW designers stop to realize the overall importance of their work as it concerns the battle fronts thousands of miles away. The fact is often overlooked that it is not only through the dramatic exploits of Flying Fortresses, tanks, jeeps and other front-line equipment that victories are being won, but also by the attention that has been given to minute details of design of these and other war machines. Standardization is one such detail that is playing an essential role.

To display originality the engineer's impulse is to design components of a special nature, in many cases failing to consider the utilization of parts that are standard and commercially available. It is fortunate for the country that during the war, however, the idea of the designer in using nonstandard parts for reasons of his own—or sometimes as "the easiest way out"—has been discarded. In its stead, the typical chief engineer's instructions to "use standard parts and stock sizes of materials wherever possible—special sizes only when absolutely necessary" is being much more implicitly followed.

Indicative of the tremendous advantages resulting from this procedure is the recent statement of an army member of the aeronautical board: "If nonstandard parts practices prevailed in aircraft manufacture, it is likely that a sizable percentage of our planes now flying would be on the ground for want of some small part with an odd-size thread!"

Not the least important among other advantages of specifying standard parts arises from the need to utilize to the full the current manufacturing facilities of the country, as brought out in the leading article in this issue. Companies that specialize in production of components for machines are in a far better position to turn out large quantities of parts and to handle rush orders than are the machine builders themselves. And such specialization, in the majority of cases, offers promise of superior quality and performance besides leaving the designer free to concentrate on other phases of his work.

Special design of complete equipment is essential to overcome the best the enemy can produce—standardization of details is just as necessary to facilitate production and to "keep 'em flying".

L. E. Jermy

Calculating Pulleys for Step-Cone Drives

By Paul Grodzinski

CORRECT belt tension when using any step of a cone-pulley drive depends on accurate determination of the pulley diameters to give the same theoretical belt length. While the equation for belt length is relatively simple to use, the inverse problem of finding pulley diameters when given center distance, belt length and speed ratio is more involved. The accompanying chart, Page 138, enables this calculation to be made with ease and accuracy.

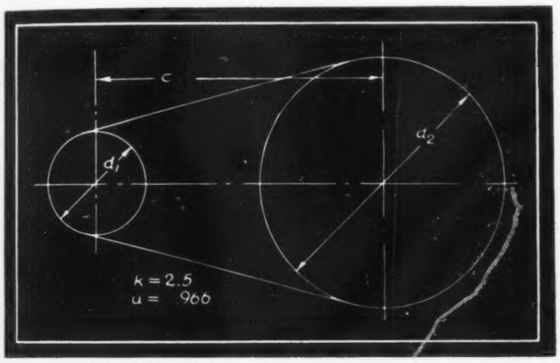
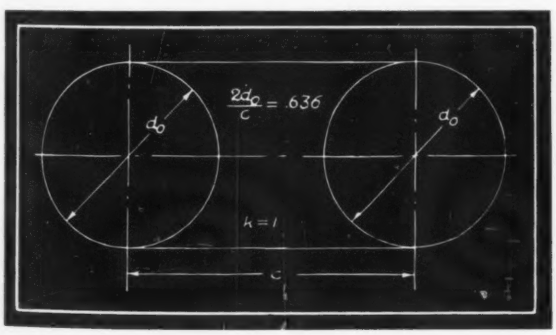


Fig. 1—Open belt drive layout for determining theoretical belt length

Fig. 2—Pulley sizes for one-to-one speed ratio are used in calculating diameters for other ratios



Basis of the chart is the well-known approximate equation for belt length on an open drive (see *Fig.* 1), which is sufficiently accurate for all practical purposes:

$$l=2c+\frac{\pi(d_2+d_1)}{2}+\frac{(d_2-d_1)^2}{4c} \qquad (1)$$

where l = belt length, inches; c = center distance, inches; d_1 = small pulley diameter, inches; and d_2 = large pulley diameter, inches. When greater accuracy is desired d_1 and d_2 may be replaced by (d_1+t) and (d_2+t), where t is belt thickness.

Introducing the speed ratio, k, which is equal to the diameter ratio d_2/d_1 and eliminating one of the pulley diameters:

$$l=2c+\frac{\pi d_1}{2}(k+1)+\frac{d_1^2}{4c}(k-1)^2 \qquad (2)$$

For the simple case of equal pulleys, *Fig.* 2, $d_1=d_2=d_0$ say, this equation becomes:

$$l=2c+\pi d_0 \qquad (3)$$

Equating the values of l in Equations 2 and 3:

$$\pi d_0=\frac{\pi d_1}{2}(k+1)+\frac{d_1^2}{4c}(k-1)^2 \qquad (4)$$

For any desired value of d_2/d_1 this quadratic equation may be solved in the usual manner. However, the resulting expression is clumsy and a quicker solution results from the following approach. As a first approximation the last term in Equation 4 may be ignored temporarily, giving as a solution:

$$d_1=\frac{2d_0}{k+1} \qquad (5)$$

Introduction of a correction factor u makes this a more exact relation:

$$d_1=\frac{2d_0u}{k+1} \qquad (6)$$

The correct value of u is given by substituting the value of d_1 from Equation 6 into Equation 4 which be-

comes, after simplification:

$$\pi = \pi u + \left(\frac{d_0}{c}\right)\frac{(k-1)^2}{(k+1)^2}u^2 \quad \ldots\ldots\ldots\ldots\ldots\ldots (7)$$

Solution of this equation is most easily accomplished with the aid of the chart, *Fig.* 3, which gives values of the correction factor u as a function of the speed ratio k for usual values of d_0/c. The basic equations for solving all problems with the aid of the diagram are summarized for convenience as follows:

$$l = 2c + \pi d_0$$

$$d_0 = \frac{l - 2c}{\pi}$$

$$d_1 = \frac{2d_0 u}{(k+1)}$$

$$d_2 = kd_1$$

It is worth noting that the value of u on the chart also indicates directly the accuracy of the common assumption that step-cone pulleys are correct if the sum of the diameters is constant. The percent error in this assumption is therefore $100(1—u)$.

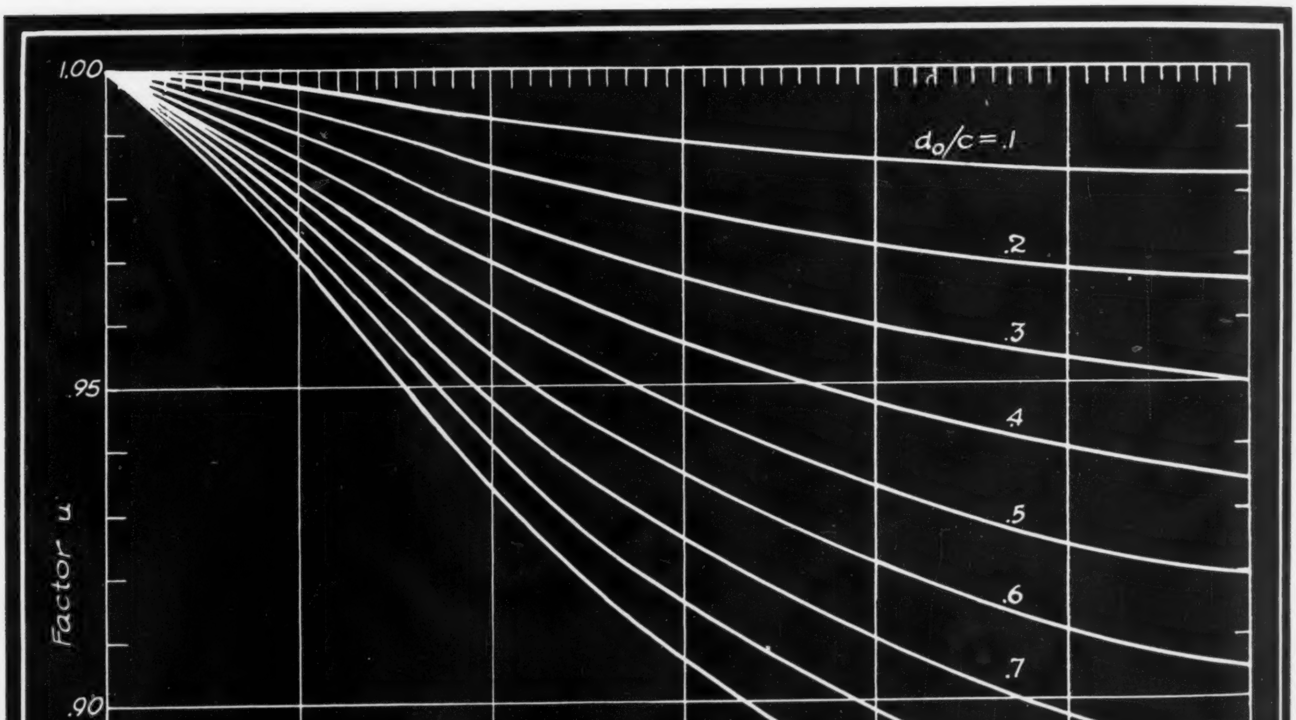
How To Use the Chart

EXAMPLE: A belt 60 inches long is to connect two shafts with a center distance of 15 inches, the speed ratio being 2.5. It is required to find the proper pulley diameters.

From Equation 3 the pulley diameter for a speed ratio of 1 may be found: $d_0 = (l—2c)/\pi = (60—30)/\pi = 9.55$. Then the ratio $d_0/c = 9.55/15 = .636$. With this value of d_0/c, and k equal to 2.5, the value of the correction factor u is found from the chart, *Fig.* 3, to be .966.

Diameter of the smaller pulley, from Equation 6, is therefore $d_1 = 2d_0u/(k+1) = (2)\ (9.55)\ (.966)/(2.5+1) = 5.26$ inches, and the larger pulley diameter is $d_2 = kd_1 = (2.5)\ (5.26) = 13.15$ inches.

Fig. 3—Chart shows correction factors for calculating pulley sizes at any step of a cone-pulley drive



PROFESSIONAL VIEWPOINTS

". . . published useful articles"

To the Editor:

At various times during the past few years, Machine Design has published some very interesting and useful articles on springs by A. M. Wahl. We are wondering if these articles could be had in pamphlet form or otherwise as they are very useful to refer to when problems in spring design arise.

—A. F. Milbrath
Vice-President & Chief Engineer
Wisconsin Motor Corp.

An authoritative book is now under preparation by Dr. Wahl on the subject of spring design which will be available later in the year. We will be glad to apprise Machine Design's readers of the date of publication through later announcements in the magazine.—Ed.

". . . chart seems complicated"

To the Editor:

In Machine Design for May, page 120, there is a chart for frequency of vibrations that seems to the writer unnecessarily complicated. Omitting the constant as not affecting the form, and transforming the equation, there results

$$f^2=k\left(\frac{1}{I_1}+\frac{1}{I_2}\right)$$

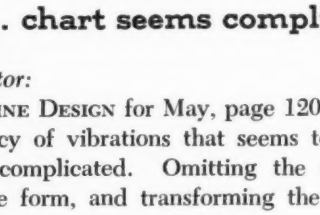
Letting

$$\frac{1}{I_1}+\frac{1}{I_2}=\frac{1}{M}$$

then

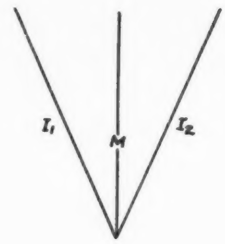
$$f^2=\frac{k}{M} \quad \text{or} \quad k=f^2M$$

which is represented by a Z-chart with scales as shown below:

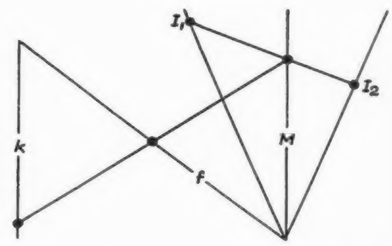


The second equation above is represented by this "V"

form of chart, illustrated below:



Combining by superposing the turning scale M and showing also two computing secants as an example,



In this, all the scales except f are proportionally graduated and M is not graduated.

—CARL P. NACHOD
Nachod & United States Signal Co.

". . . scale depends on range"

To the Editor:

There are so many ways of constructing a chart to represent a given relation between a certain number of variables, that it is always debatable which one to choose. The choice usually depends upon the range of each of the variables and the accuracy desired within certain ranges.

In the present case, if the range of f and k had been small, the type of chart suggested by Mr. Nachod would have been selected, but the value of f running from 5 to 500 makes it difficult to construct a natural square scale and have the same degree of accuracy over the entire range. This is the reason why a logarithmic scale was used in the prepared diagram. I believe that the additional step necessary in reading the chart is well warranted under these conditions.

—R. D. DOUGLASS
Massachusetts Institute of Technology

". . trying to find answer"

To the Editor:

Your article "When Helical Springs Buckle" in the May issue of MACHINE DESIGN has interested the writer very much, because for years he has been trying to find this answer.

We at once checked the formula on an air valve spring which we are using now. It buckles slightly and can be used on the engine but, before we go into production, the spring should be redesigned to eliminate the buckling. Characteristics of this spring are

NICKEL AIDS THE AUTOMOTIVE INDUSTRY

to KEEP 'EM ROLLING!

Using ingenuity and "know-how" born of long experience, automotive engineers designed the phenomenally successful transport equipment that now speeds the United Nations on the road to Victory.

Built to take punishment far above peacetime requirements, these specialized military vehicles are being produced in quantity by the mass-production methods that have amazed the world. From North Africa to the South Pacific, these trucks, jeeps, tanks and half-tracks have repeatedly met wartime demands for stepped-up performance.

This kind of engineering-thinking pioneered the application of Nickel alloyed materials. Now, when uninterrupted operation is so vitally important, the continued and widespread use of Nickel is clear evidence of its many advantages.

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For years the technical staffs of International Nickel have been privileged to cooperate with automotive engineers and production men . . . men whose work is now so necessary to the Nation. Counsel, and printed data about the selection, fabrication and heat treatment of ferrous and non-ferrous metals is available upon request.

* *Nickel* *

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THE INTERNATIONAL NICKEL COMPANY, INC., *67 Wall St., New York, N. Y.*

$C_k = 33.5$ pounds per inch deflection
$d = .244$-inch wire
$n = 13\frac{1}{4}$ active coils, ends closed and ground flat
Mean diameter $= 2.244$ inches
$r = 1.122$ inches
$l_o/r = 11/1.122 = 9.8$
Factor $C_B = .53$

then

$$P_{cr} = .53 \times 11 \times 33.5 = 195 \text{ pounds.}$$

Still, when the spring is compressed into precompressed condition (static) to 6½-inch length, the load is only 151 pounds and a slight buckling occurs.

—H. Schreck,
Fairbanks, Morse & Co.

". . . is necessary to allow margin"

To the Editor:

With reference to Mr. Jackson's letter, he has calculated the buckling load of the spring by assuming built-in ends (i.e. the upper curve of *Fig.* 5 of the writer's article). In most practical springs, however, the condition of built-in ends is not perfectly realized and for this reason the actual buckling load will be below that calculated on this basis.

The writer believes that the difference between the calculated value of 195 and 151 pounds, at which slight buckling was observed is probably due primarily to the fact that the ends of the spring are not perfectly fixed as assumed in the calculation. Even if the spring is compressed between heavy plates which remain absolutely parallel, the presence of the end turns introduces a certain amount of flexibility as well as eccentricity, which in turn reduces the buckling load below that calculated on the fixed-end assumption. Any flexibility present in the supporting structure will also add to the present due to the end turns.

Although available test data indicate that the formulas given in the paper are sufficiently accurate for most purposes provided the end conditions are known, it must be admitted that such test data are comparatively meagre. For this reason, further tests on actual springs to determine the effects of deviations from the ideal conditions would be highly desirable. Such deviations include not only lack of complete fixity of the ends, but also initial variations in spring dimensions, eccentricity of loading, lack of parallelism of ends and other factors. When the results of such tests are available, the designer will be able to form a better judgment as to the required margin of safety between the actual and calculated load for the so-called fixed-end condition. In the absence of experimental data, use of the lower curve of *Fig.* 5 (of the writer's paper) based on the assumption of hinged ends should yield safe values of the buckling load, but is probably unnecessarily conservative in most instances.

The writer appreciates the example cited by Mr. Schreck, which shows that in design it is necessary to allow some margin between the actual load and calculated buckling load if fixed-end conditions are assumed.

—A. M. Wahl
Westinghouse Elec. & Mfg. Co.

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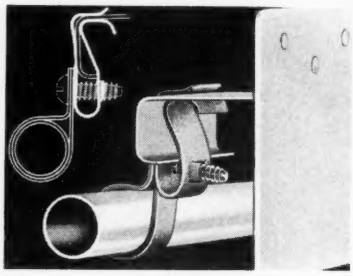
New PARTS AND MATERIALS

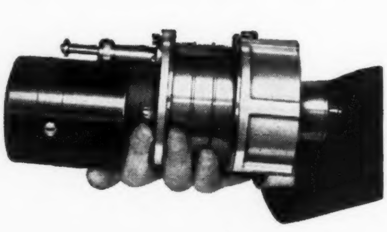
Aircraft Nut for Rolled Sections



AN ENTIRELY new Speed Nut has been developed by Tinnerman Products Inc., 2085 Fulton road, Cleveland, for a more rapid assembly of conduit, piping and wire harnesses in aircraft. This nut, Type 6337, is of special aircraft spring steel with a zinc metal spray finish, and is designed to quickly snap around rolled sections or stringers, eliminating need of drilling holes which weaken the structure. Legs are forced inward to give a firm spring tension grip as the screw in the nut is tightened. It is made for 8Z and 10Z sheet metal screws and for various "Z" stringers.

Midget Variable-Speed Drive

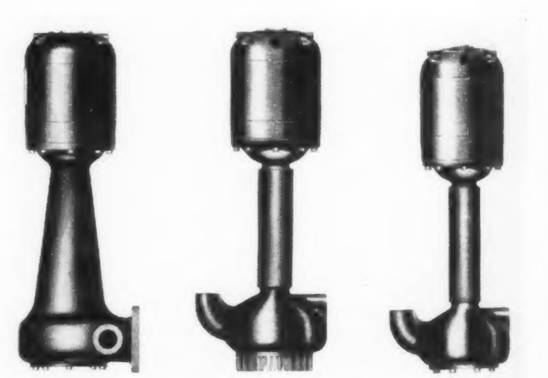
FOR use in the "flea-power" range, Graham Transmissions Inc., 2706 North Teutonia avenue, Milwaukee, 6, Wis., is offering its new Midget variable-speed drive. Weighing only 8 pounds complete with motor this drive, known as Model 15, is applied to motors of not more than 1/15-horsepower. Identical in design principle with the larger units manufactured by the company, using fully



standard ball bearings, the variable-speed transmission is available in torques ranging from 4 inch-pounds at maximum forward speed (for 3600 revolutions per minute input) to 8 inch-pounds in reverse; or from 1 inch-pound at maximum forward speed (for 1800 revolutions per minute input) to 2 inch-pounds at reverse. Output speed range is from 700 revolutions per minute to zero, or if it is desired to reverse the output shaft without reversing the motor, the range is from 400 revolutions per minute forward to 400 reverse. Instantaneous and shockless reversal is due to the low inertia of parts (output shaft and gear only), the high-speed parts connected to the motor continuing to turn at all times at constant speed. The new model is adaptable to various controls. Overall length including input and output shaft extensions is 7⅝, depth with adjustable mounting legs 4⅜ inches, and height from base to top of motor 4⅞ inches.

Versatile Coolant Pumps

EACH available in seventeen standard sizes, three new coolant pumps have been announced by Pioneer Pump & Mfg. Co., Detroit. Since constant flow is assured, the possibility of causing scrap due to pump failure is greatly reduced. Another feature of the pumps is their versatility of application, permitting interchangeability with other makes of pumps. Model VBD in this group



has three outlets to permit piping on either right or left side, or back into coolant sump through intake bracket. Another model, known as VA, is used where submersion in coolant sump is more practical or less objectionable than outside mounting. Model VC is designed for external use where a tank cannot be utilized efficiently.

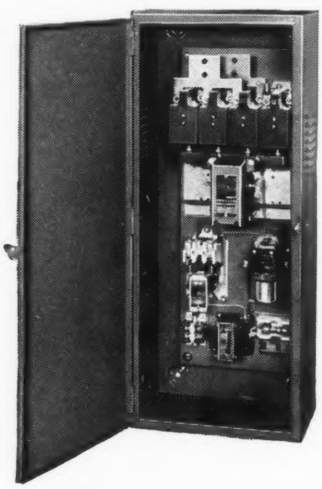
Lightweight Switch for Aircraft

DESIGNED specially for aircraft applications where space is limited and where severe vibration conditions are encountered, General Electric Co.'s new small, lightweight, dust-tight switch has a contact mechanism of the snap-action, double-break type. For use in a wide range of ambient temperatures — from 95 degrees Cent. to —40 degrees Cent.—the switch is corrosion-proof, meeting 200-hour salt water tests, and is suitable for altitudes from sea level to 40,000 feet. It is

a spring-return, plunger-operated type switch with a 7/32-inch overtravel, and can be mounted either on the cover or opposite side, thus facilitating operation of plunger from either right or left. Available in three different contact arrangements—single-circuit, normally open; single-circuit, normally closed; and two-circuit, normally open or normally closed—the switch is furnished with either die-cast zinc or die-cast aluminum housings and with either solder lug connections or, in the single-circuit forms, with an AN connector insert built in the AN threaded nipple of the housings. For use in locations where mud and sand are encountered, it is also available with a rubber boot mounted over the plunger mechanism.

Safety Control for Welders

ANNOUNCED by the Industrial Controller division of Square D Co., Milwaukee, is a safety panel which eliminates the hazard created by high, open-circuit voltages of transformer type alternating-current arc welders. By automatically disconnecting the welder transformer after the arc is broken, operator injury is eliminated. Transformer is automatically reconnected the instant op-



erator touches the electrode to the work. Saving in power is effected when a number of welders are used on a single power system because the welding transformer primary circuit is connected to the power line only while the arc is held. The panels can be used with any make or model of alternating current transformer type arc welder. External connections are simple and internal connections of the welder need not be disturbed.

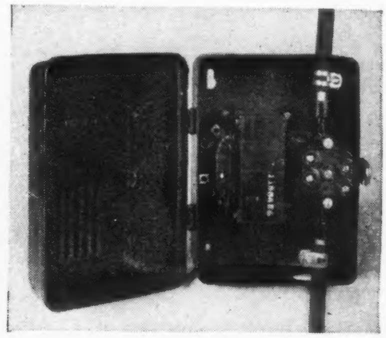
Electro-Thermostatic Flow Switch

DETECTING the lack of proper cooling water required for adequate cooling of ignitron tubes in resistance welding and other applications, a new electro-thermostatic flow switch has been made available by Westinghouse Electric & Mfg. Co. Designed for use where a liquid for cooling is employed or where a smooth positive acting flow switch is required, it consists of a 100-watt transformer having a one-turn secondary short

(*Continued on Page* 154)

(*Continued from Page* 148)

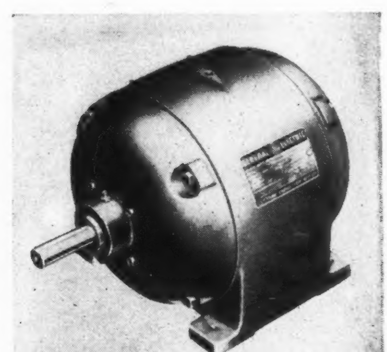
circuited through a piece of high resistance stainless steel tubing through which flows the water from the ignitron tubes. A normally closed, two-pole thermostatic switch is used, it is available in either a 1½ or 3½ gallons per minute size in both a 25 or 50/60-volt rating. Other



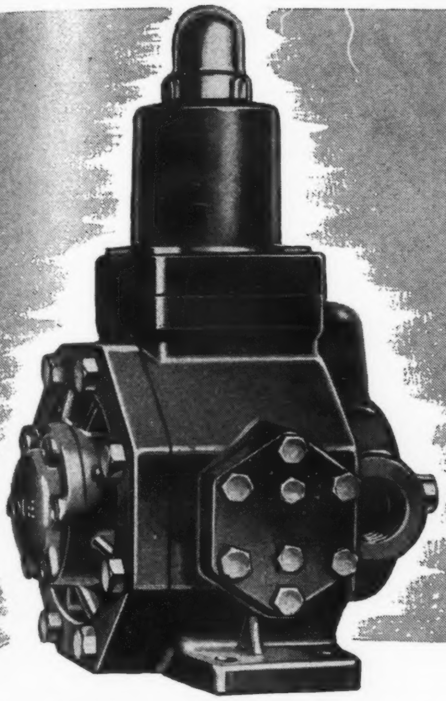
voltage ratings may be obtained by supplying an autotransformer. To insure against excessive tightening of switch elements, the switch is furnished with a spring mounting, with or without enclosing steel cabinet. Space required for mounting is approximately 7 x 9 x 5 inches.

Totally Enclosed Motors

RECENTLY added to the group of Tri-Clad motors furnished by General Electric Co. is a new line of totally enclosed motors which are available in both the polyphase, 60-cycle, induction type and the single-phase, 60-cycle, capacitor type. These motors are especially designed for use under conditions where abrasives, chemicals, rain, snow and excessive dirt are encountered. Hav-



ing all the basic features of the Tri-Clad group, the polyphase motors are available in frame sizes 203 to 225 and include ½, ¾ and 1-horsepower motors at 900 revolutions per minute; ¾, 1 and 1½ horsepower at 1200 revolutions per minute; 1, 1½ and 2 horsepower at 1800 revolutions per minute; and 1½ and 2 horsepower at 3600 revolutions per minute. The single-phase type motors are furnished in frame sizes 203 and 204, and include ¾ horsepower at 1200 revolutions per minute; 1 and 1½ horsepower at 1800 revolutions per minute; and 1½ and 2 horsepower at 3600 revolutions



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Racine Hydraulic Pumps are the most modern "source of force" for clamping, forming, moulding, feeding, bending, lifting and many other metal working operations.

Racine Pumps operate at reduced horsepower, because the flow of oil is automatically varied according to the amount of oil required to do the job. Made in three sizes—0-12, 20 and 30 gal. per min. at 50 to 1000 lbs. pressure per sq. in. Equipped with standard automatic pressure controls; or with Solenoid, Lever or Hydraulic two-pressure control; or with Handwheel or Lever for manual volume control.

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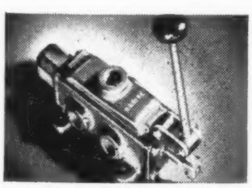
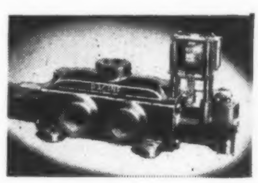


With Balanced Piston—exclusive "Sleeve Type" construction. Piston always in alignment, cannot sag. Unique porting design reduces hydraulic shock. Also special porting arrangement to fit your job. Made in 3/8" to 1 1/2" standard pipe sizes with mechanical, electrical or manual operating devices.



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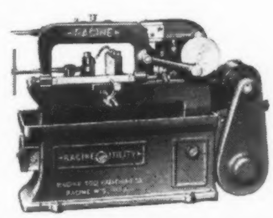

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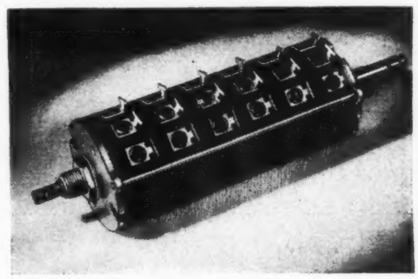
per minute. Mounting dimensions are interchangeable with Tri-Clad open motors of the same ratings. Triple protection, against physical damage, electrical breakdown and normal operating wear and tear, is provided in the motors. Exceptional resistance to corrosion and blows because of the cast-iron frame end shields and conduit boxes is also a feature. Liquids are prevented from seeping into the motor by leads permanently encased in compound in a cast-in pocket in the stator frame. Further protection is provided by a rotating labyrinth seal, preventing dirt, oil or water from entering the bearing.

Tandem Controls Announced

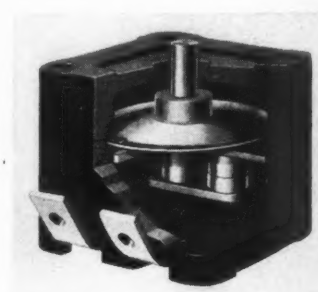
DEVELOPED to meet certain radio and electronic requirements calling for single control of several circuits, the No. 42 series control introduced by Clarostat Mfg. Co. Inc., 285-7 North Sixth street, Brooklyn, can control a plurality of circuits—up to two dozen. The new design of case for each unit permits nesting and locking into a compact stack. Metal end disks and tie rods



hold cases together and provide further rigidity. The single shaft passes through the locks with each rotor in the stack. All units of the control pass through the same degree of rotation as the single shaft is rotated. Individual units can be of any standard resistance, taper, taps and hop-offs to meet individual circuit requirements. Since the number of sections and values vary, these series controls are made for special orders only.

Sealed Aircraft Switches

SEALED in cases to protect their contacts against hazards of dirt, dust, sand and oil, Nos. A3 and A5 single-pile, single-throw switches offered by Allied Control Co., 2 East End avenue, New York, are finding acceptance for electrical control



units for the newest types of aircraft and ground equipment, especially where climatic conditions cause failure of ordinary types of equipment. Operating characteristics of the switches are: For A3 type, normally closed, double-break; for A5, normally open double-break; contact ratings, noninductive, 5 amperes at 12 and 24 volts

A Tribute for Outstanding Gear Production

For outstanding gear production—gears long noted for their dependability, accuracy and adherence to close tolerances—Alloy Steel Gear & Pinion Company have been awarded the Army-Navy "E" for Excellence. Ten active years in the gear business have given us a background of experience that makes it possible for us to produce even better gears now—faster than ever before.

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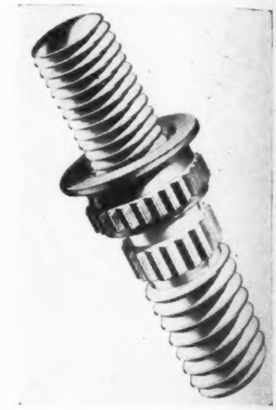
direct current and 110 volts alternating current; operating pressure, 1½ to 3½ pounds; plunger travel, .006 to .012-inch; overtravel, .05 to .07-inch at maximum pressure; vibration, 10g for either horizontal or vertical positions; weight, 5 ounces; dimensions, 1 15/16 x 1 15/16 x 1 19/32 inches.

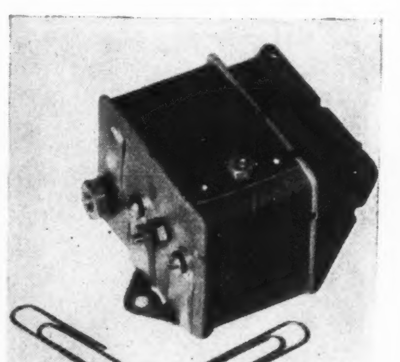
Locking System for Fastenings



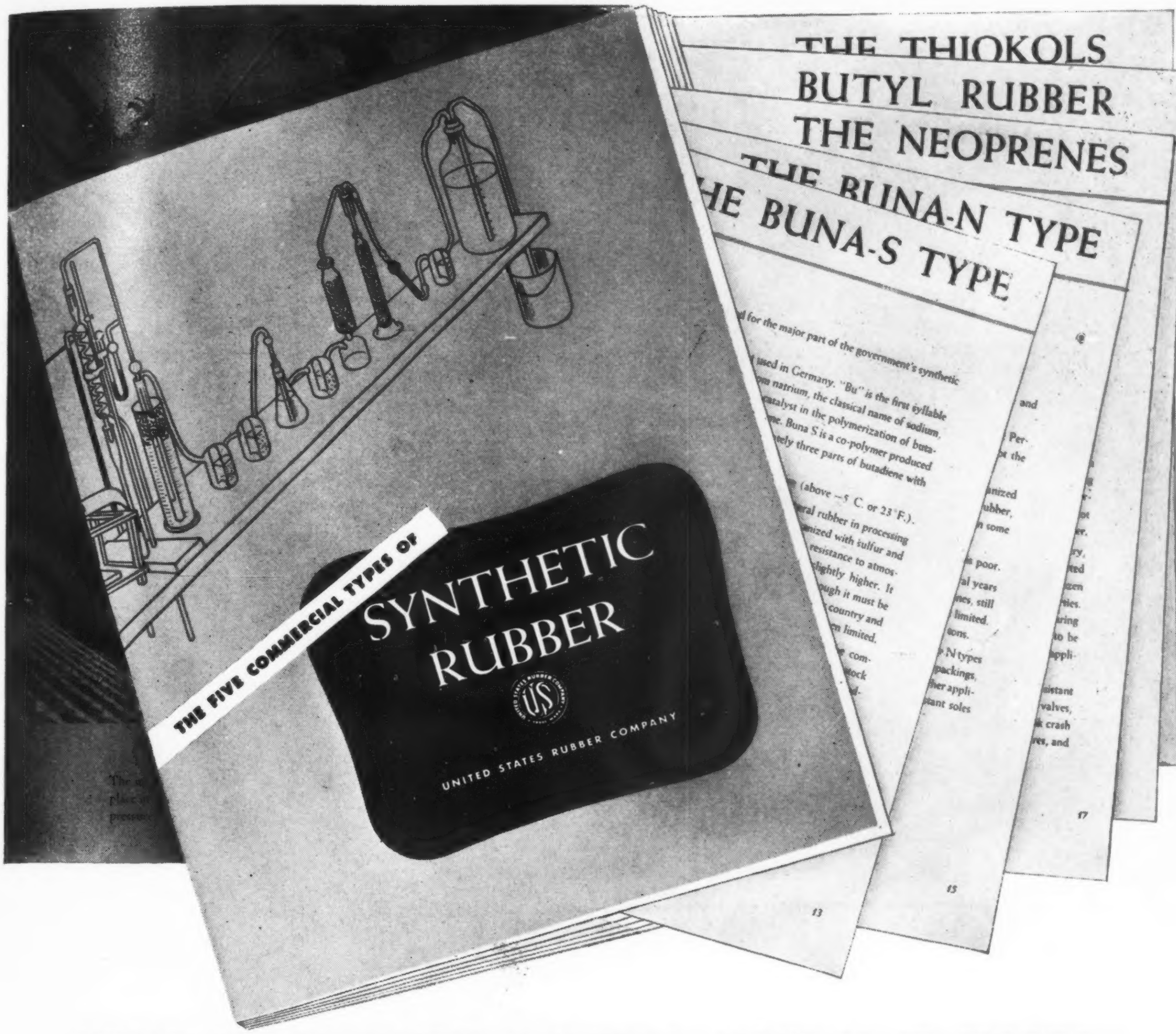
PROBLEMS of permanent installation of fastenings have been solved by Jose Rosan, inventor of the Rosan locking system for threaded inserts and studs, which are now being manufactured by Bardwell & McAlister, Hollywood, Calif. The principle of the new locking system is simple. A locking ring serrated inside and out engages its inner teeth, where installed, with a serrated collar on the insert or stud. Outer teeth broach their way into the softer material when struck. The insert or stud thus becomes an integral part of the softer material and the serrations prevent the ring from turning. Uses of this system and types of studs and inserts manufactured under it are numerous. One application is for fastening plastic noses on bombers; another for electrical panels. A spark plug type insert is also offered which can be inserted into an aluminum engine block, while a cone sealed type of insert is used for gases and liquids under pressure.

Lightweight Aircraft Contactor

ESPECIALLY effective for controlling solenoids and small motors in aircraft, a new contactor has been made available by General Electric Co. for use where



reliable performance under adverse conditions is needed. Conforming with the specifications of the Army Air Force, it is furnished in two sizes—50 and 100 amperes. The size of the 50-ampere type is 2 5/16 x 2 inches, weight



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Having worked in the field of synthetic rubber for more than twenty years, we know what each of the five types will do; what chemicals such as sulfur, carbon-black, or ultra-accelerators must be added, and how to compound them. We work with all five types; use the type available that is best suited for the purpose.

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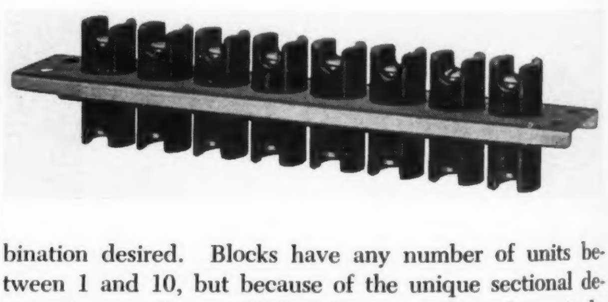
4¾ ounces; the 100-ampere type is 2¾ x 2⅝ inches and weighs 11 ounces. The unusual light weight of the contactor is due largely to its balanced armature construction. It can be mounted in any position on either a metal or nonmetallic base, and is suitable for use at altitudes from sea level to 40,000 feet.

Nonpriority Wrinkle Finishes

DEVELOPED by Maas & Waldstein Co., 438 Riverside Drive, Newark, N. J., a new line of wrinkle finishes known as "Victory" finishes have been announced. The new finish closely resembles standard wrinkle finishes, manufacture of which is restricted for a few special applications and soon is to be prohibited entirely due to Chinawood oil shortage. Hard, durable coatings are formed by the new finish, which covers rough metal surfaces effectively in a single coat. It is applied in regular wrinkle patterns by the same methods as these and is obtainable in a full range of colors.

Multiple-Terminal Block

FOR subpanel and chassis construction, with feed-through terminals, a new multiple-terminal block has been introduced by Curtis Development & Mfg. Co., 1 North Crawford avenue, Chicago. Designed to meet today's demands of electronic and electrical design requiring external terminals, the terminal block consists of individual feed-through terminals, mounted in bakelite, which are permanently held in a metal strip in any com-

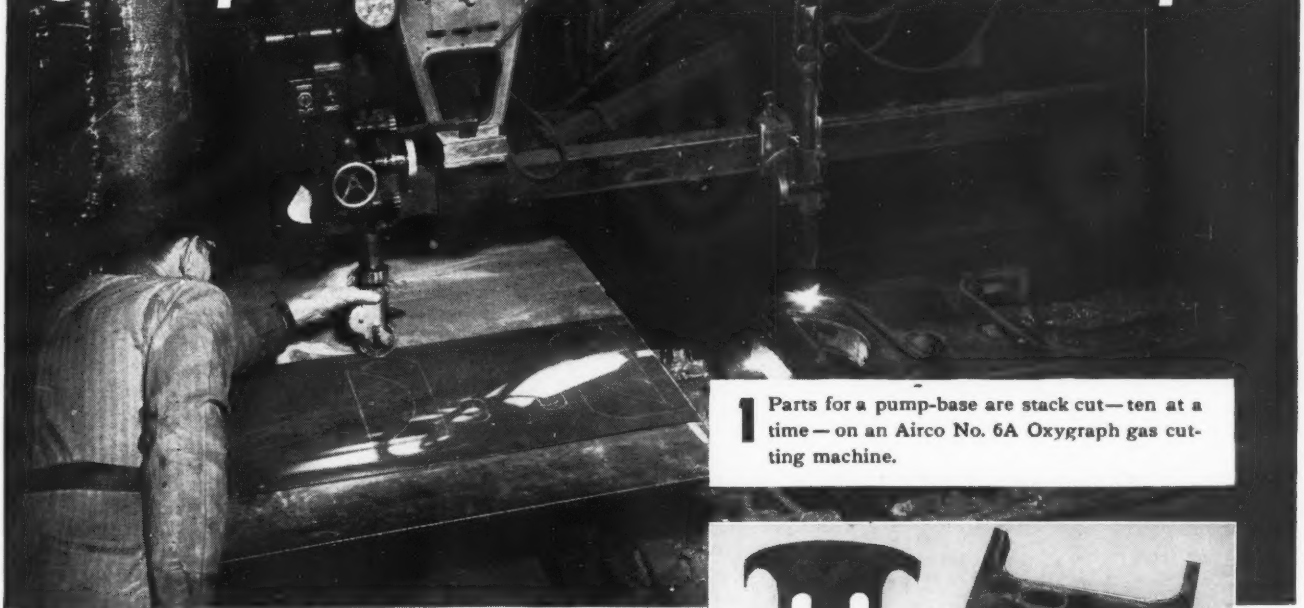


bination desired. Blocks have any number of units between 1 and 10, but because of the unique sectional design, they can be supplied with any number of terminals. Terminals have ample clearances and leakage distances for circuits carrying up to 300 volts, 20 amperes. Center to center distance between terminals is ⅝-inch. No. 8 screws are used for securing connection, and in the mounting holes at each end of the terminal base.

Material for Nameplates

FOR instruction plates, dials, etc., Colonial Brass Co., Middleboro, Mass., has developed its material known as Coballoy as a substitute for critical metals such as steel, brass, zinc and aluminum in conjunction with the Conservation Branch of the War Production Board and the Armed Services. Coballoy can be furnished in raised or sunken letters and the quantities can be large or small. It is available in two hardnesses, soft and hard, the soft

3 steps to a fabrication triumph



1 Parts for a pump-base are stack cut—ten at a time—on an Airco No. 6A Oxygraph gas cutting machine.

2 Here are the flame-cut shell parts, each 1-4" thick, before and after separation. These parts are next rolled to the required curvature.

3 Uniting the separate parts into a sturdy product by electric arc welding. Two flame-cut and rolled shell parts make up each pump base.

Featuring:

FLAME CUTTING and ARC WELDING

Simple, sturdy, and economical—these traits distinguish this pump base which was designed and built by Avery and Saul Company, Boston, Massachusetts.

Taking full advantage of the flexibility offered by flame-cutting and arc welding, this manufacturer achieved both speed and economy in producing these bases.

This modern metal fabrication team offers advantages that can simplify many metal working problems. For full details and engineering aid on versatile oxyacetylene and electric arc processes, get in touch with your nearest Air Reduction office.

AIR REDUCTION

AIRCO

60 E. 42nd STREET, NEW YORK 17, N. Y.

In Texas: Magnolia Airco Gas Products Co.

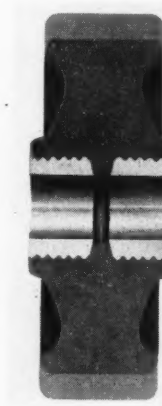
FOR VICTORY BUY UNITED STATES WAR SAVINGS BONDS STAMPS

IDLE CYLINDERS ARE PRODUCTION SLACKERS: KEEP 'EM ROLLING FOR VICTORY!

being used where instruction plates must be bent to curved surfaces and the hard for the flat surfaces. Melting point of the material is in the neighborhood of 460 degrees Fahr., and the material has less shrinkage than brass or bronze.

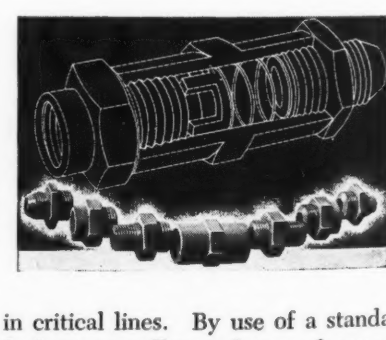
Wheel with Armored Metal Hub

RESINOID wheels with armored metal hubs have been made available by Rapids-Standard Co. Inc., Grand Rapids, Mich., for practical application to power-hauled industrial trailers, or for hand-operated trucks. The tread of the new wheel is ABK laminated plastic fabric of the type used in ship stern tube bearings and steel rolling mill bearings. It is bonded under high heat and pressure to a shock resisting core. Hub is reinforced with armored metal inserts to insure bearing fit and shock resistance, and to eliminate excessive side thrust wear. The wheels will not harm floors or chip and will carry heavy loads. They are highly resistant to oils and greases and can withstand exposure to mild acid and alkaline solutions, as well as operating satisfactorily under temperatures up to 200 degrees Fahr. The two sizes available are 10 x 3 inches and 12 x 3½ inches, both with or without roller bearings. The wheel is also furnished in diameters ranging from 4 to 12 inches, without the armored-metal hub.

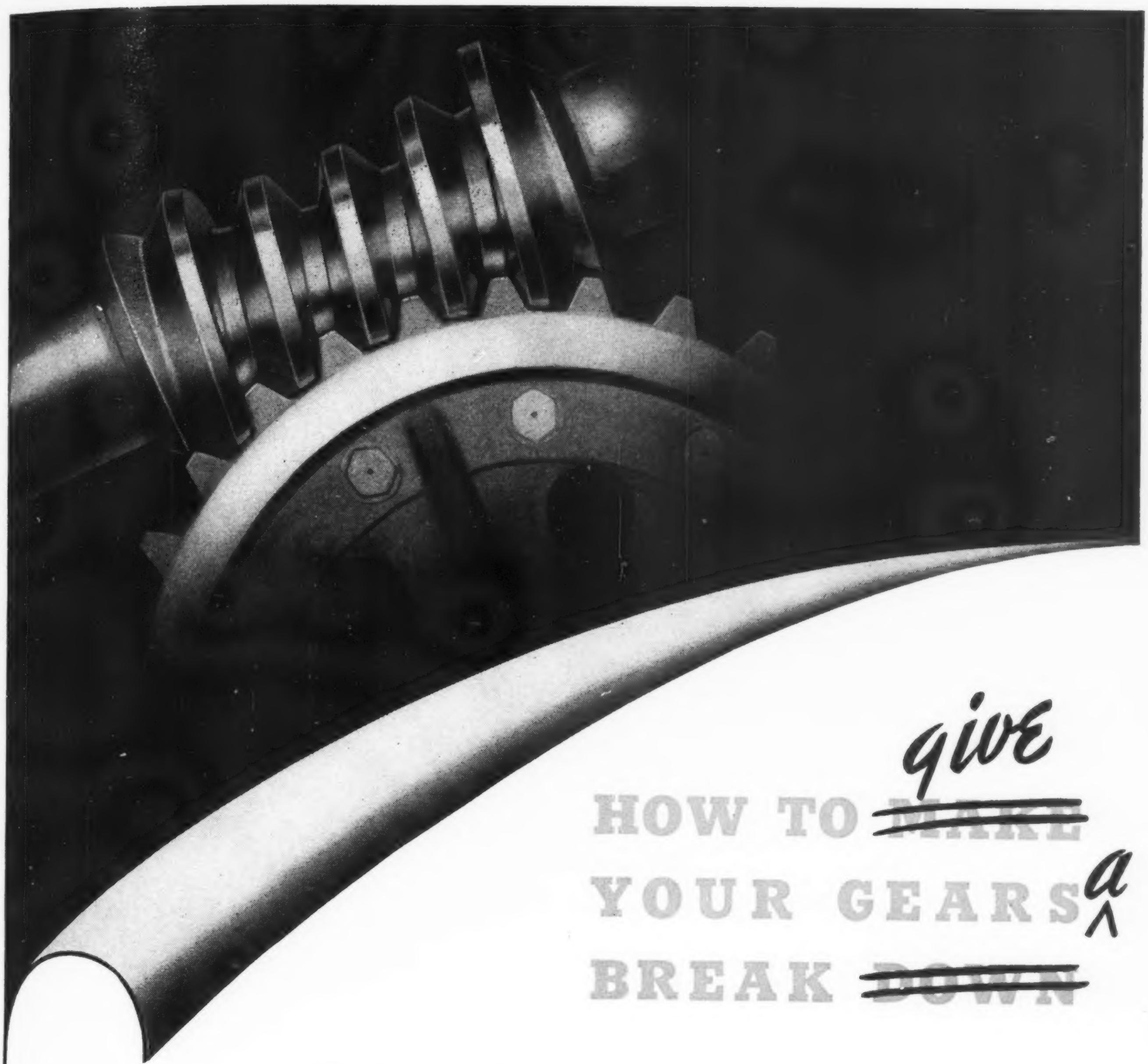


Hydraulic Check Valves

AVAILABLE in standard sizes, either with light phenolic poppets or with single or double steel ball checks, the new universal check valves offered by American Screw Products, 7000 Avalon boulevard, Los Angeles, are made to operate in any position, are unaffected by vibration, variable pressure or acceleration and will give



protection in critical lines. By use of a standard body of heat-treated aluminum alloy and two of any of the three types of adapters, six basic combinations of connections are possible with flared tubing or internal or external pipe threads. Each adapter is provided with a seat for a



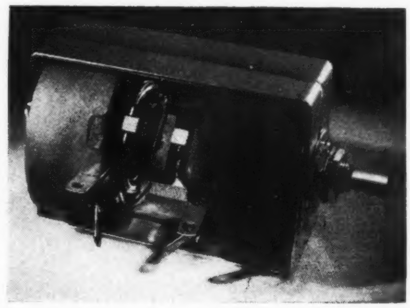
phenolic poppet. By reversing the poppet position or the entire valve assembly, direction of fluid flow can be changed, making twelve possible combinations. There is a 16 per cent reduction in weight in the new check valves for aircraft hydraulics.

Synthetic Rubber Parts

TO SUIT the particular requirements of engineers and designers, Lights Inc., 1111 South Fremont avenue, Alhambra, Calif., is offering resilient parts of all types, made to order. The custom-molded synthetic rubber of which parts are made is compounded to give resistance to oil, gasoline, chemicals and aging. It is available in molded, fabricated and bonded construction in tough, resilient parts.

Tandem Power Rheostats

TANDEM power rheostat assemblies of two or more sections have been developed by Clarostat Mfg. Co. Inc., 285-7 North Sixth street, Brooklyn. The new assemblies are made up of two 25-watt or two 50-watt rheostats rigidly coupled together and held in a metal cradle. Usual one-hole mounting and locking-projection

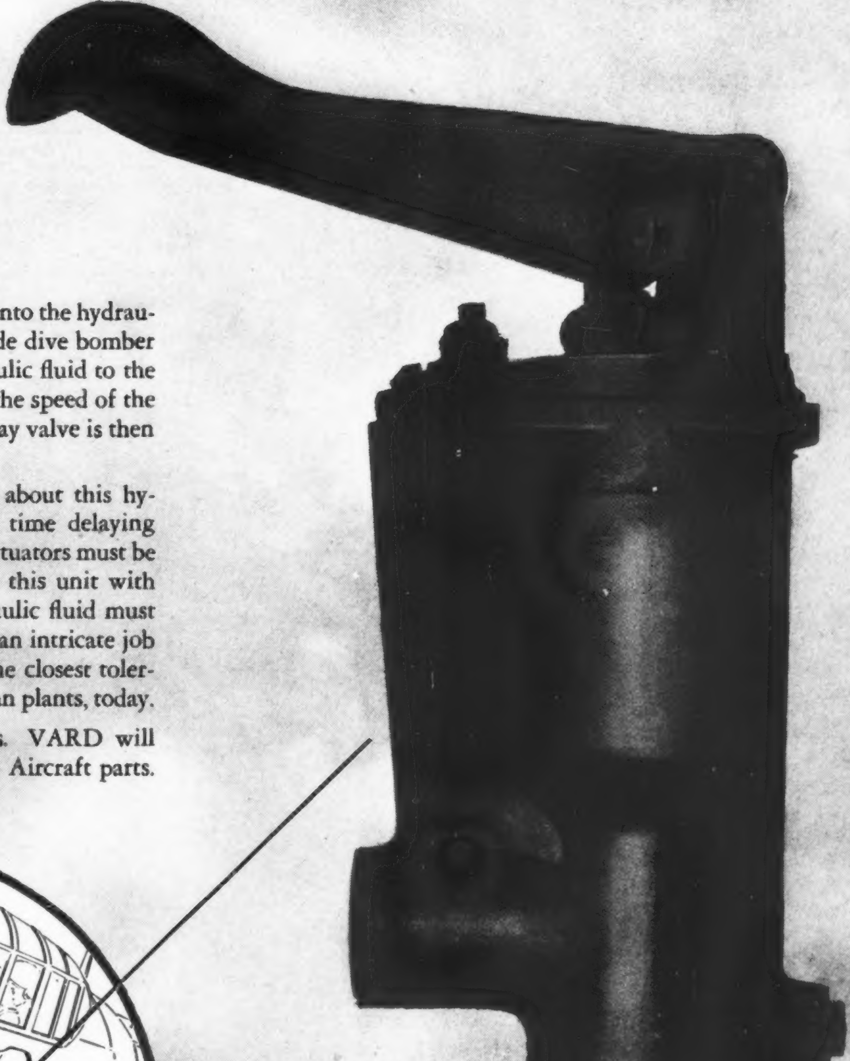


features have been retained. The individual units can be of any standard resistance value, taper, tap and hop-off, and all go through the same degree of rotation as the single shaft is turned. The rheostats are made on special orders because of the wide choice of resistance values and other factors. Units are all fully insulated from each other and from the ground.

Fluorescent Aircraft Signals

OPERATING by fluorescent reflection of "black light", a small signal announced by Littelfuse Inc., 4757 Ravenswood avenue, Chicago, works in daylight, under "black light" and no light, and is used in aircraft wherever a signal light is required. Indication is entirely by reflected light and radio activity. A radium-active fluorescent paint used on the indicator shows signals in total darkness, and a saving in current is effected as it uses only 1.5 watts as against the approximate 4½ watts now used. There is no blurr occasioned by transmitted light, and clear visibility of signals is dependably effected inasmuch as the pilot does not have to adjust his eyes

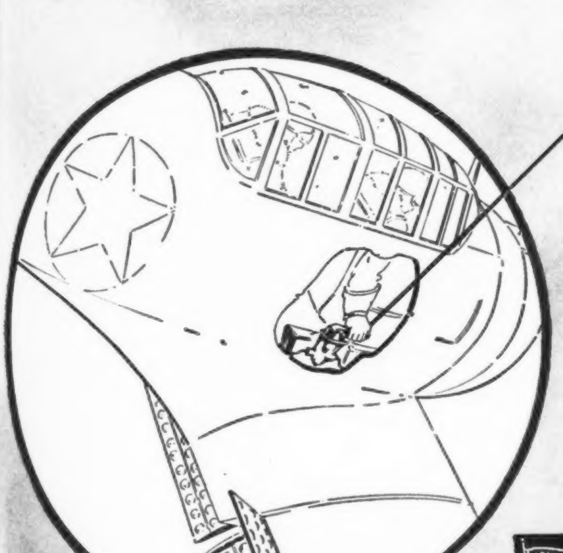
SKY BRAKE



This Time Delay Valve fits into the hydraulic system of an American made dive bomber and controls the flow of hydraulic fluid to the wing flaps. These flaps brake the speed of the plane in descent. The time delay valve is then a brake control.

The distinguishing feature about this hydraulic unit is of course the time delaying element. Ordinary hydraulic actuators must be precision made, certainly, but this unit with measured control of the hydraulic fluid must be precision plus. In fact, it is an intricate job of machining right down to the closest tolerances being worked in American plants, today.

VARD makes these valves. VARD will always be making the finest in Aircraft parts.



In times of Peace–Prepare for War In time of War Prepare for Peace! All Americans must realize that war industries with depleted reserves, sinking funds and profits, can never rally to meet the Crash of Cancelled Contracts We must have reserves to meet the storm–and Win the Peace

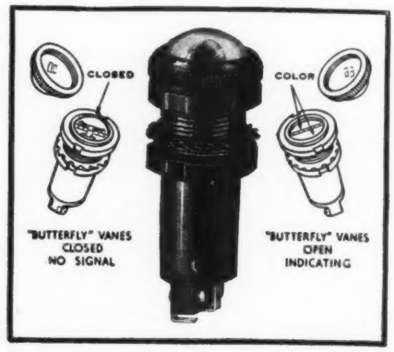
ARMY E NAVY

VARD INC.

Thread & Plug Gag s • Snap Gages • Plain Tapered Ring Gages • Bench Model External Comparators • Dividing Machines • Precision Ground Optical Lenses & Filters • High Fidelity Mirrors

PASADENA, CALIFORNIA, U.S.A.

to the reflected light. The signal becomes correspondingly brighter in strong light. Another improvement is its nonshatterable transparent plastic cap which withstands the most severe test of shock or explosion and permits free penetration by ultraviolet rays. The body of the indicator



houses a solenoid, the armature of which is connected with the "butterfly" indication vanes by a simple lever hookup. The fluorescent butterfly opens instantly to show signals, reflecting the proper indicating light. Butterflies are furnished in red, amber and green. When not indicating the signal is black. Length overall is 2-5/32 inches, and unit is made for mounting in panels up to ⅜-inch thickness.

Engineering Dept. Equipment

Prevents "Blueprint Rash"

WITH the use of a new development called "Hi-Tense Concentrate," recently introduced by Wisconsin Pharmacal Co., Milwaukee, "blueprint rash" can be prevented. The new developer is nonirritating to hands immersed in it and has the additional advantage of producing sharper blueprints, free from brown spots, as well as being completely noncorrosive to tank linings and nonstaining if spilled on clothing. Furnished in highly concentrated form, a pint bottle can be diluted with water in a ratio of 1:320—making 40 gallons of the solution.

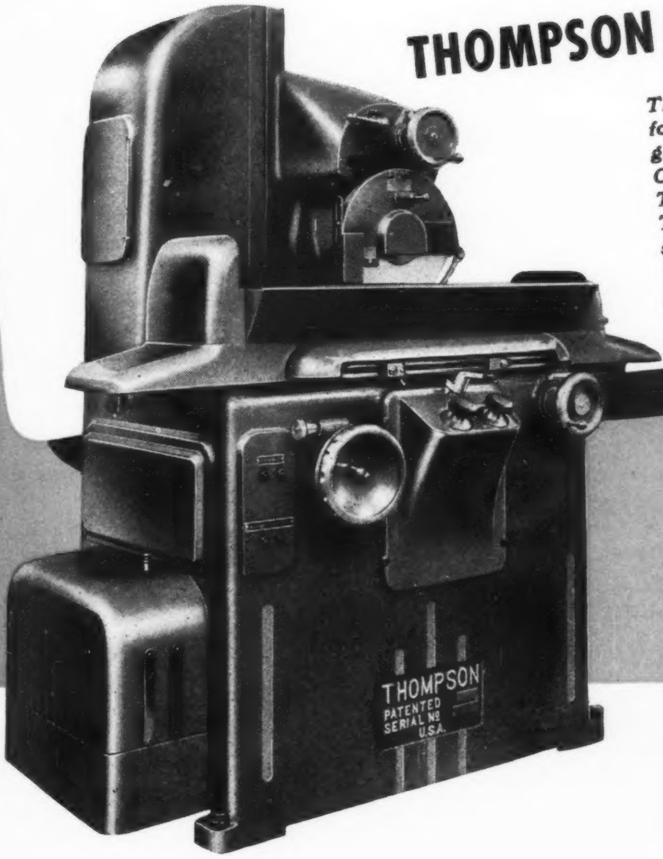
Vest-Pocket Slide Rule

OFFERED by Tavella Sales Co., 25 West Broadway, New York, is a Mascot vest-pocket slide rule which gives the square, the square root, the logarithm and reciprocals of all numbers. The "trig" scale gives the sines and tangents of numbers. These scales are circular, therefore endless, and answers are never off scale. They are on disks 2¾ inches in diameter and length of multiplication-division scale is 6.3 inches as against 5 inches on "A" scale of the regular 10-inch slide rule. Made of white celluloid with fine black graduation, the rule is easily readable and can be dropped, trampled on and immersed in liquids without its usefulness being impaired. It is grease and waterproof, and can be easily washed with soap and water.

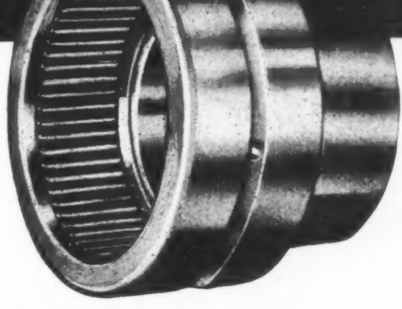
Meeting the Precision Requirements of

THOMPSON SURFACE GRINDERS

Thompson Surface Grinders are well-known for their high-speed, smooth, precision grinding. Contributing to this efficiency, Orange Roller Bushings are used in the Table Hand Feed Unit of the Type F Thompson Hydraulic Surface Grinders, shown herewith. After 4 years of continuous use, the Thompson Grinder Company reports, "Excellent performance"!



ORANGE ROLLER BUSHINGS



Many machinery manufacturers select Orange Roller Bushings for their great load carrying capacity in small space. However, in other applications requiring utmost precision (such as above), the unusually even, quiet, long-life operation of Orange Roller Bushings proves equally valuable.

This smooth action is the result of closer running clearances, achieved by holding roller clearances to a minimum through a rigid standardization of size, thus preventing rollers from running tangent to race.

Whether your product requires ruggedness — precision — or *both combined* . . . you have the answer in Orange Roller Bushings. Made in a complete range of types and sizes. Our engineers are glad to discuss any bearing application.

Mail Coupon for Engineering Data

Orange Roller Bearing Co., Inc., MD
Orange, N. J.
Please send me your Roller Bushing Data Book

Name.................................. Title..........

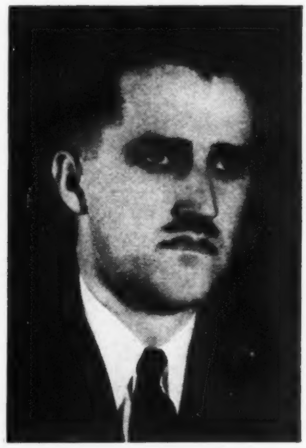
Company..

Address ....

City........................... State..............

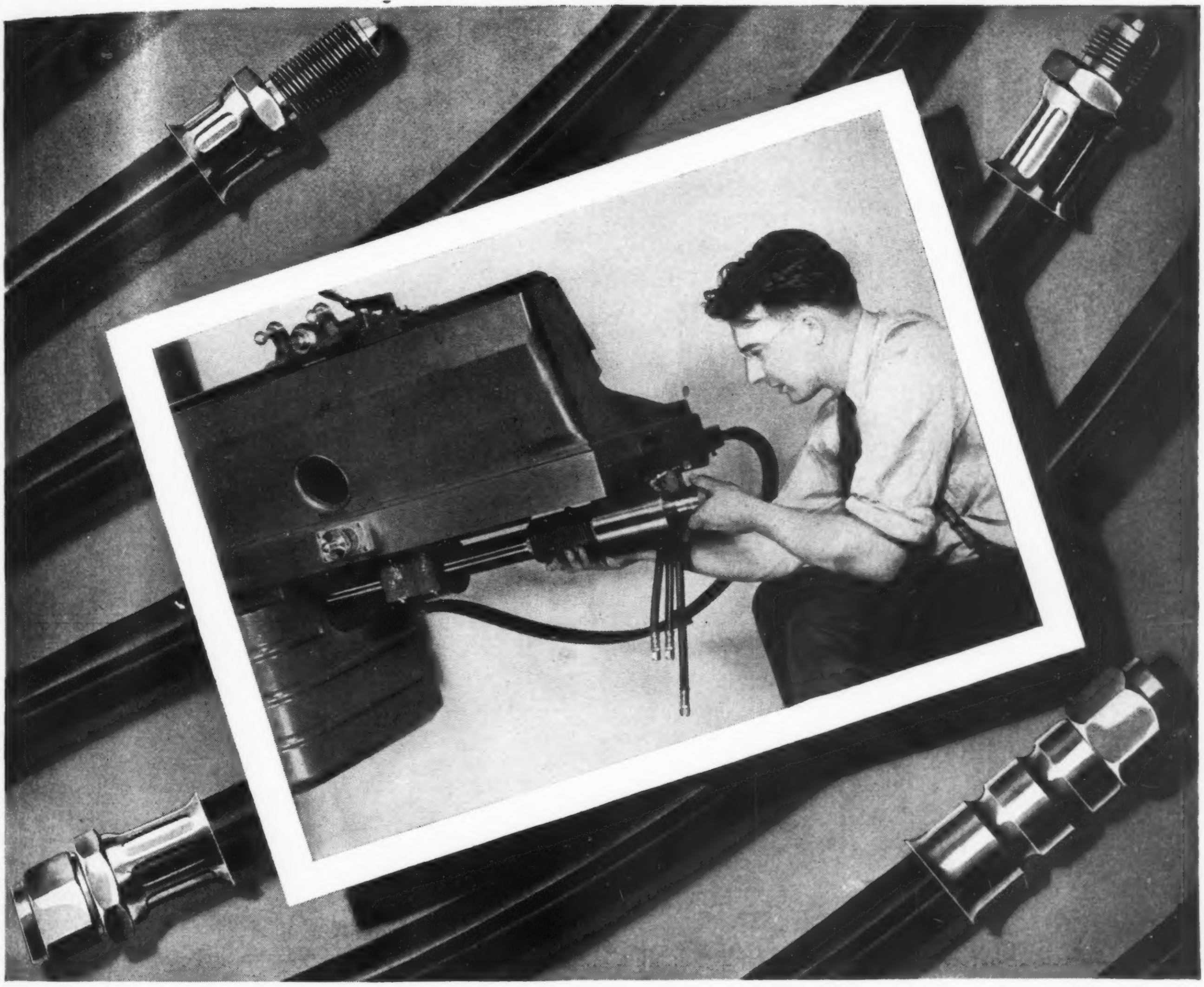
ORANGE ROLLER BEARING CO., INC.
ORANGE • NEW JERSEY

MEN OF MACHINES



FOR many years chief engineer of the Peerless Pump division, Food Machinery Corp., James M. Hait was recently appointed general manager of the procurement and engineering division. He is credited with many of the "firsts" in pump design, and during the past year has been working diligently in the development of amphibian tanks for the government, with outstanding results. Consequently the company established the new division of which he is now the head. His services in the design and development of amphibian tanks, and such other equipment as he has been called upon to design will aid materially in helping to win the war. Mr. Hait is a 1928 mechanical engineering graduate of Rensselaer Polytechnic institute. Prior to his connection with the Food Machinery Corp. he worked for one year for Emsco Aero Engine Co., and four years for Peerless Pump Co.

RESIGNATION of Everett Chapman who since 1936 has been president of Lukenweld Inc. has been announced by Lukens Steel Co., of which Lukenweld is a subsidiary. Mr. Chapman will establish his own consulting engineering business, numbering Lukens Steel Co. among his clients. Pending election of a successor, administration will be in charge of G. Donald Spackman, recently elected vice president at Lukens. Mr. Chapman, a native of Detroit, graduated from the University of Michigan in 1923 with a degree in electrical engineering. After postgrad-



"Your hose ended all valve clogging."

SAYS PROMINENT MACHINE TOOL BUILDER

Clogging of hydraulic valve mechanisms is no new problem but it can be a serious and costly one. Vital equipment pulled down for repairs—key engineers and mechanics tied up on service work.

Engineers at one of this country's leading machine tool manufacturing plants were wide awake to the seriousness of such a situation. They knew that such clogging sometimes resulted from gummy materials extracted by the hydraulic fluids from flexible hose assemblies.

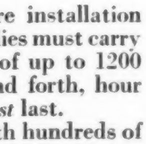
After thorough investigation, Resistoflex hose was chosen for this installation, and what could have caused a troublesome service problem was avoided.

This is no low pressure installation either. These hose assemblies must carry an intermittent pressure of up to 1200 lbs., be slammed back and forth, hour after hour...and they *must* last.

Our files are bulging with hundreds of interesting applications that may prove helpful in solving some of your flexible hose problems. Write us about them today, or...

SEND FOR CATALOG

Write for the Resistoflex Industrial Catalog. On your company stationery, please.



RESISTOFLEX FEATURES:

STRONGER—Resistoflex hose assemblies out-point all similar lines in tensile strength.

PERMANENT, FULL FLOW—Chemically inert, glass-smooth inner surface provides permanent free flow—eliminates turbulence and skin friction.

NON-CLOGGING—Resistoflex hose assemblies do not gum, never clog hydraulic or lubrication systems, diesel injector nozzles or other fine orifices.

FLEXIBLE, VIBRATION PROOF—Tens of thousands of simultaneous flexings and twistings have no effect on Resistoflex lines.

HOSE AND HOSE ASSEMBLIES FOR HYDRAULIC OILS AND VACUUM, FUELS AND LUBRICANTS, ORGANIC SOLVENTS, PAINTS AND LACQUERS, THINNERS, REFRIGERANT, NATURAL AND MANUFACTURED GASES—LABORATORY TUBING—DIPPED AND MOLDED GOODS—COATING COMPOSITIONS, GLOVES AND PROTECTIVE GARMENTS.

RESISTOFLEX

RESISTOFLEX CORPORATION, BELLEVILLE, NEW JERSEY

uate work, he also obtained his master of science degree. He taught electrical engineering at Purdue University and later served as an experimental engineer with Lincoln Electric Co. As director of development and research he joined Lukenweld and in 1934 was elected vice president. Two years later he was made president. He is nationally recognized as an authority on welding, and is the author of many publications on various phases of the subject.

RECENT addition of E. L. Potter to the executive and technical personnel of French & Hecht Inc. promises much for the success of the company's war program as well as development and postwar planning. Mr. Potter who has been elected vice president of the company brings to his post a background of valuable experience which eminently suits his new capacity as executive head of the engineering and sales activity of the organization. His first experience in the field of experimental engineering was gained in the McKeen Motor Car plant of the Union Pacific Shops. After being in the World War, he returned and obtained his mechanical engineering degree at the University of Michigan in 1924. For a number of years he was in charge of research and experimental work for the Hupp Motor Co. during the period marked by the development of the company's first 6-cylinder motors. He then became connected with Gabriel Co., developing a hydraulic spring control important in modern automotive practice. After some private engineering consultant work, he joined the Houde Engineering division of Houdaille-Hershey Corp., concentrating upon the development of hydraulic spring control again. He was then promoted to sales manager of the company, and substantially contributed to the advancement of the hydraulic control, now standard on all streamline trains.

A. W. Herrington, past president of the Society of Automotive Engineers, received the honorary degree of mechanical engineer at the seventy-first annual commencement exercises of Stevens Institute of Technology.

William Waddell has joined Studebaker Corp., South Bend, Ind., as special engineer. He had been chief draftsman at the Lycoming division of The Aviation Corp., Williamsport, Pa.

Richard C. Gazley, formerly engineering manager of Porterfield Aircraft Co., has become associated with the Cleveland Pneumatic Tool Co.

William A. Cresswell Jr., analytical engineer, Ranger Aircraft Engines, division of Fairchild Airplane &

Engine Corp., Farmingdale, L. I., N. Y., has been transferred to junior project engineer in the stress and vibration department.

♦

ALVIN J. KORNBLUM is now project engineer with Hughes Aircraft Co., Armament division, Hollywood, Calif. Previously he had been assistant project engineer, Vega Aircraft Corp.

♦

GUNNAR JENSEN has been transferred to the Racine, Wis. office of the Walker Mfg. Co. of Wisconsin, as chief engineer. He formerly had been chief engineer of this company's Jackson, Mich., division.

♦

SERGE L. CROWELL has resigned as designing engineer of Babcock & Wilcox Co., New York City, to accept a position as helicopter designing engineer at Sikorsky Aircraft, Division of United Aircraft Corp., Bridgeport, Conn.

♦

PETER F. ROSSMAN has recently been appointed general manager of a new division known as the Development Division of Curtiss-Wright Corp. He formerly was chief of development research in the Airplane division, Research Laboratory and came to Curtiss-Wright from Packard Motor Car Corp. where he held executive positions in connection with design. He also was assistant to the director of military engineering and technical assistant.

♦

T. M. CUMMINGS, vice president since Progressive Welder Co., Detroit, was founded, has been made executive vice president and general manager. Heretofore he has been in charge of manufacturing and engineering development as well as sales.

♦

DR. CHARLES M. SLACK, well known in the field of electronics research and for his contributions to the development of an ultra high-speed X-ray machine, has been named assistant director of research at the Westinghouse Lamp Division, Bloomfield, N. J.

♦

WELLWOOD E. BEALL has been elected vice president in charge of engineering of Boeing Aircraft Co. He formerly was chief engineer. EDWARD C. WELLS is the new chief engineer of the company, replacing Mr. Beall.

♦

A. E. BEDELL has recently been appointed chief engineer of Graver Tank & Mfg. Co. Inc., in charge of all engineering and development. Prior to his new connection, Mr. Bedell had been associated with Max E. Miller & Co. Inc., and for 14 years was in charge of engineering for that organization.

♦

ALEXANDER KARTVELI, vice president and chief engineer of the Republic Aviation Corp., has been honored by the Temple university with a degree of Doctor of Science. The degree is being conferred on Mr. Kartveli for designing the P-47 Thunderbolt high-altitude fighter plane.

♦

ASHLEY C. HEWITT has recently been appointed chief engineer of the engine division, Radioplane Co., Van

Nuys, Calif. He was previously connected with Kinner Motors Inc., Glendale, Calif., as experimental test engineer.

♦

ROBERT PATRICK VAIL is head of mechanical engineering at the Pantex Ordnance Plant, St. Francis, Tex. Prior to this appointment he was assistant professor of mechanical engineering and coordinator of civilian pilot training at the Texas Technological College, Engineering Division.

♦

HOWARD BARLOW has been elected chairman of the Society of Aeronautical Weight Engineers. He is head of the aeronautical engineering department, Texas A & M College.

♦

PAUL W. EELLS has been promoted from executive engineer to vice president in charge of engineering, LeRoi Co., West Allis, Wis.

♦

JOHN O. KOBZINA JR., formerly diesel development engineer at International Harvester Co., Chicago, has joined the General Machiney Corp., Hamilton, O., as project engineer.

♦

EDWARD J. NESBITT has joined the structures section of the Helicopter engineering department, Sikorsky Aircraft, Stratford, Conn.

♦

WILLIAM G. ANDERSON, previously an instructor in mechanical engineering at the Newark College of Engineering, is with the National Advisory Committee for Aeronautics, Cleveland Airport, Cleveland.

♦

T. R. THOREN, who has been working in the production of pumps, Thompson Products Inc., has been designated as chief development engineer.

♦

EDMOND E. BISSON, associate mechanical engineer, Aircraft Engine Research laboratory, National Advisory Committee for Aeronautics, has been transferred from Langley Field, Hampton, Va., to Cleveland.

♦

ROY M. SMITH has succeeded J. D. Wood, resigned, as chief engineer of Roller-Smith Co.

♦

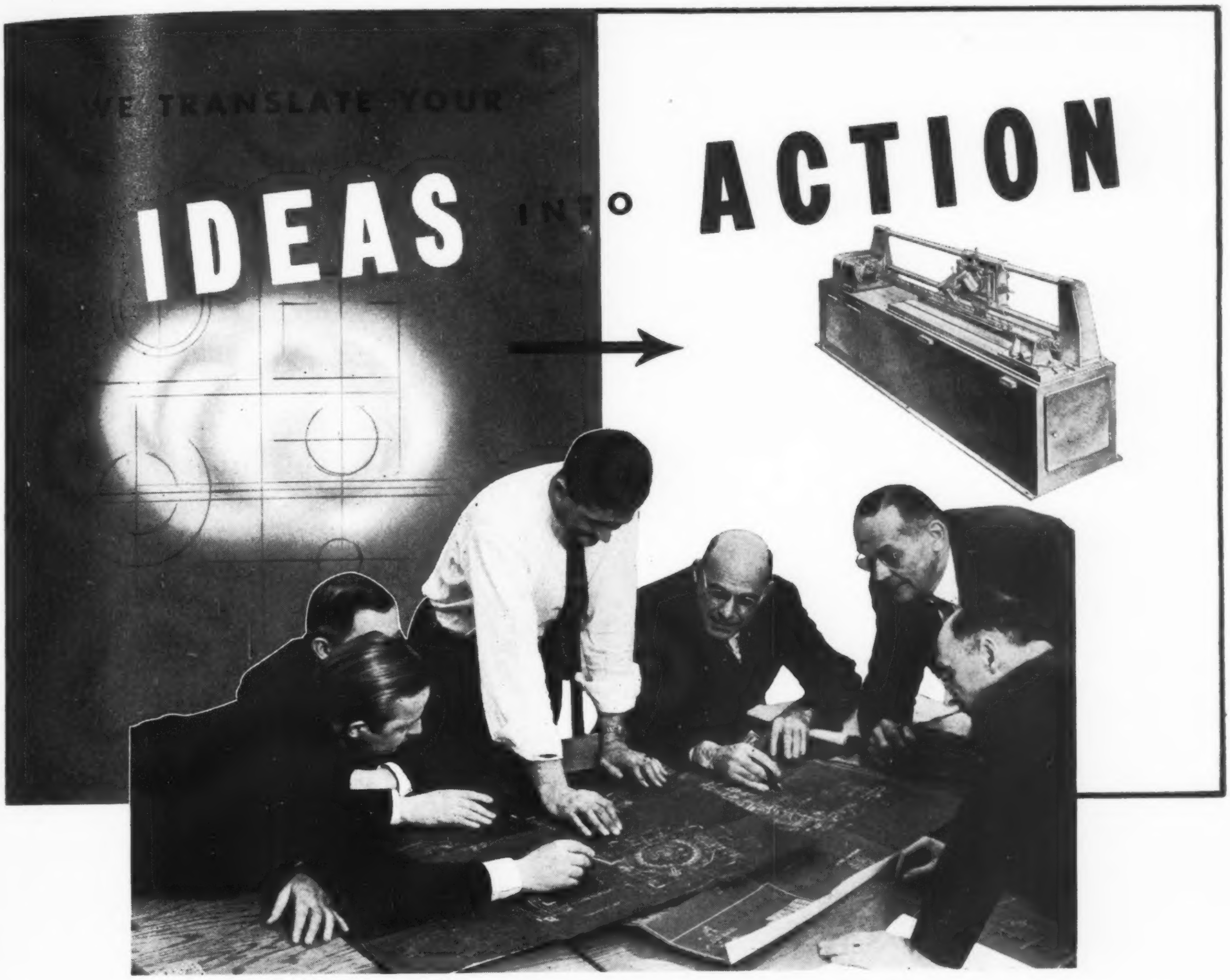
WALTER A. SEMION has been made project engineer at Higgins Aircraft Inc., New Orleans, La. His previous connection was that of assistant chief engineer of H. J. Heinz Co., Aircraft Product Division, Pittsburgh.

♦

LOUIS R. BOTSAI, manager, gearing department, Nuttall Works, Westinghouse Electric & Mfg. Co., was elected vice president of the American Gear Manufacturers association at the annual meeting.

♦

AUBREY A. ROSS, recently retired after completing 48 years' service with General Electric Co., has been designated as the second recipient of the Edward P. Connell award established by the Falk Corp. as a memorial to its late vice president. This is given in recognition of contributions to the gear industry, the gearing arts or the association.



PRODUCTION to win the war . . .

PLANNING to win a post-war market

Perhaps you have a production problem, a cost reduction problem or a new product development problem that you know could and should be solved, were it not for lack of man power or facilities to design and build the needed special production equipment. There is hardly a plant in operation today where such a situation does not exist.

We are Production Engineers . . . an organization 300 strong with the brain-power, the man-power and the production facilities to put your ideas and plans into ACTION now. We have accumulated a wealth of experience in practical fabrication of many of the new materials destined for widespread post-war application. We have designed and built the special machinery that is today helping many famous war plants produce complex components faster, more accurately and at lower unit cost. We have also designed and built the necessary dies, tools, jigs and fixtures, in some cases for an entire mass production schedule. We are equally prepared to function on the small individual production problem or a complete tooling program.

Production to win the war is still American Industry's primary job. Preparation to win peace-time markets is a management obligation. We have amply demonstrated our ability to serve in both capacities. Write or wire for consultation without obligation.

DESIGNERS ATLAS BUILDERS
TOOLS AND DIES
JIGS AND FIXTURES
SPECIAL MACHINERY

A LOAD OFF
YOUR SHOULDERS
ONTO OURS

ATLAS

METAL STAMPING COMPANY
TOOL & DESIGNING COMPANY

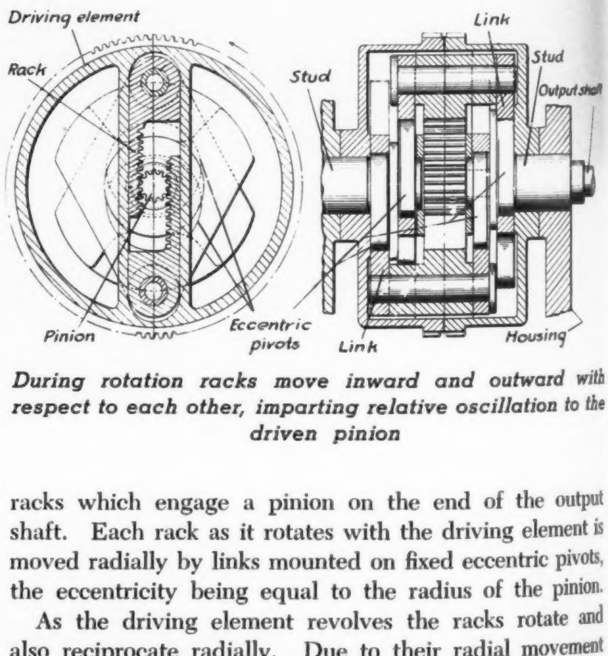
CASTOR & KENSINGTON AVENUES : : PHILADELPHIA, PA.

Noteworthy Patents

Provides Cyclic Speed Variation

Cyclic speed variation of a driven shaft with constant driving speed may be accomplished with minimum wear and vibration through the use of a mechanism covered by patent 2,314,278, recently assigned to the United Shoe Machinery Corp. During a complete revolution of the input shaft the output speed varies from zero to a maximum of twice input speed and back to zero, the average speeds being equal.

As shown in the figure the driving element is a cage formed of two cup-shaped members mounted for rotation upon studs carried by a housing. Gear teeth on the periphery enable it to be continuously driven by suitable means. A slot or guideway extending across the diameter of the driving element controls the movement of two

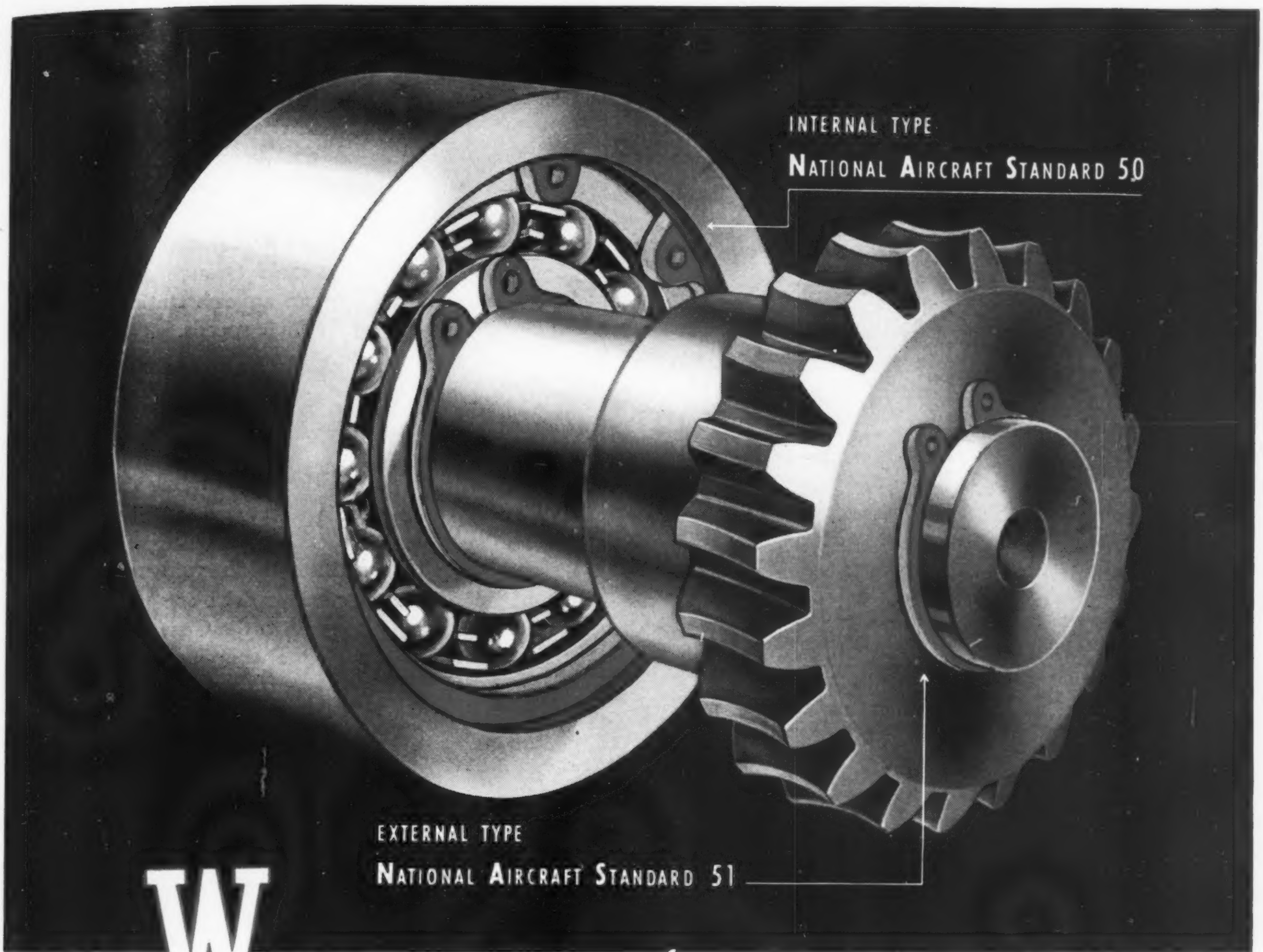


During rotation racks move inward and outward with respect to each other, imparting relative oscillation to the driven pinion

racks which engage a pinion on the end of the output shaft. Each rack as it rotates with the driving element is moved radially by links mounted on fixed eccentric pivots, the eccentricity being equal to the radius of the pinion.

As the driving element revolves the racks rotate and also reciprocate radially. Due to their radial movement the racks impart an oscillation to the output pinion, while due to the rotation of the racks the pinion also rotates at the same speed. Superposition of these two motions results in giving the output pinion the desired cyclic speed variation.

In the position shown the racks momentarily have no radial motion, being at the outward position of their travel, and the pinion speed is equal to the driving speed. After 90 degrees counterclockwise rotation the racks are moving inward at their maximum velocity and the pinion rotates at twice the driving speed due to the additional component resulting from the relative movement of the racks. After 180 degrees rotation from the position shown the racks again have no radial movement, being at



Waldes Truarc presents a significant advance in retaining rings. It spreads or contracts without distortion; always retaining its perfectly fitting circular contour. For thrust-load fixing, and shaft and housing applications, Waldes Truarc provides distinct advantages over nuts and bolts or wedges and washers . . . it reduces dimension and weight . . . saves material . . . cuts manufacturing time . . . simplifies assembly and dis-assembly. On request, we will gladly furnish samples and full data for your tests.

WALDES TRUARC RETAINING RING

U. S. Patent Re 18144

WALDES KOH-I-NOOR-INC. LONG ISLAND CITY-N.Y.

Sole Manufacturers in U.S.A.

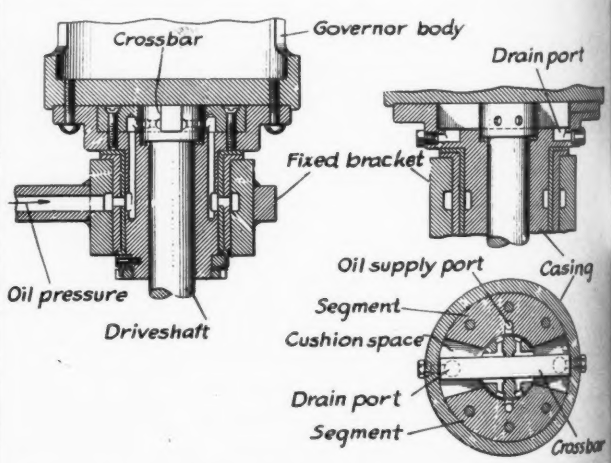
the inward position of their travel, and the pinion speed is equal to the driving speed. After further rotation to the 270-degree position the racks are moving outward at their maximum velocity, rotating the pinion in the clockwise direction with a speed which exactly offsets the driving speed, resulting in the pinion coming momentarily to rest.

By providing two oppositely disposed racks which move simultaneously inward or outward along the guideway, and by driving the racks through two sets of links on opposite sides, the parts are substantially in balance, obviating undue wear at high speeds.

Hydraulic Coupling With Positive Drive

MINOR speed fluctuations of a driving shaft often must be prevented from affecting the driven shaft, as in the case of speed-responsive governor drives. A hydraulic coupling which satisfies this requirement and at the same time provides a positive drive is covered by patent 2,307,506, recently assigned to the General Electric Co.

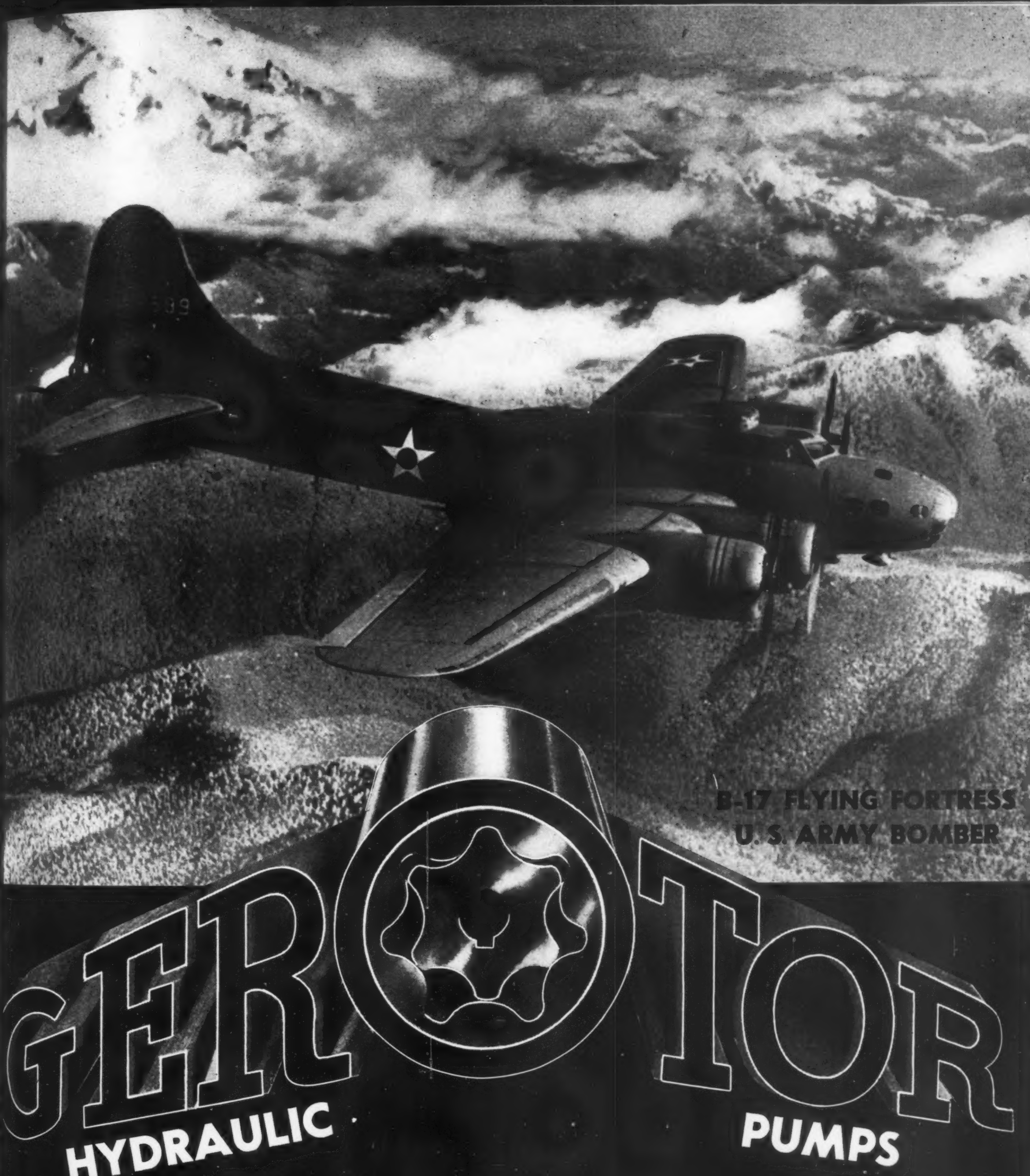
Principal elements of the coupling are illustrated in the figure, which shows an arrangement suitable for driving a governor with a vertical spindle. Upper, enlarged end of the driveshaft is provided with a transverse slot in which a crossbar slides freely, projecting beyond the head.



Oil cushion permits limited angular displacement. Pressure restores coupling to central position

The driven assembly comprises a cylindrical casing provided with a flange by which it is secured to the governor body. Casing is supported by a bearing secured to a fixed bracket, a shoulder at the upper end and a collar at the lower preventing axial movement.

Bore of the casing is enlarged at the upper end forming, with the driveshaft head, an annular recess. Two segments are attached to the casing on opposite sides of this recess, leaving spaces into which the overhanging ends of the crossbar on the driveshaft project. Spaces between the crossbar and segment faces are kept filled with oil, forming a cushion through which the driving torque is transmitted. Oil is conducted under pressure through drilled passages to the oil supply ports adjacent to the clearance spaces between the segments and the driveshaft head. With the crossbar in the central position as shown

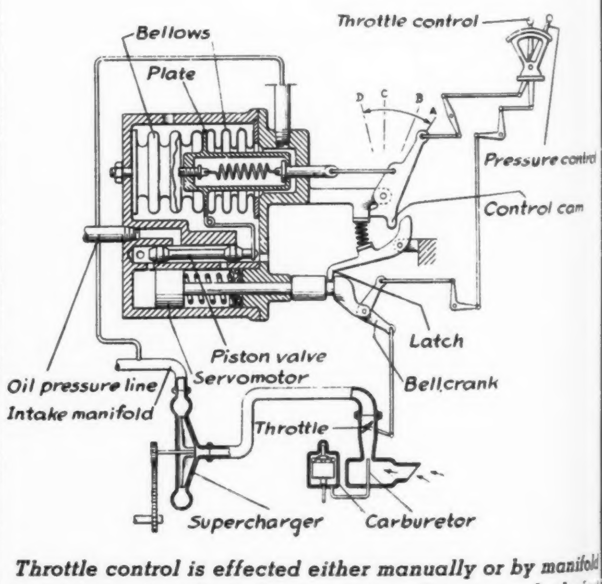

B-17 FLYING FORTRESS
U. S. ARMY BOMBER
GEROTOR
HYDRAULIC
PUMPS

in the plan view oil flows to the cushion spaces in restricted amounts through the small clearances.

If during operation the driveshaft with the crossbar is suddenly accelerated, the increased pressure ahead of the driving faces will force oil to flow through the clearance between the ends of the crossbar and the casing into the cushion spaces behind the crossbar. As a result of the angular displacement of the driving and driven elements the oil supply ports register with corresponding ports in the driveshaft head leading to the cushion spaces behind the crossbar, thus building up pressure which tends to restore the crossbar to a central position. No additional oil reaches the cushion spaces ahead of the crossbar, but if the angular displacement is sufficient the crossbar uncovers drain ports in the casing, permitting oil to leave these cushion spaces and increasing the pressure differential on the two faces of the crossbar.

Servomotor Locking Mechanism

WITH servomotor mechanisms it sometimes is desirable to make provision for locking out the automatic feature for a selected portion of the stroke where manual control is required. An example is found in the throttle control for aircraft engines in which intake manifold pressure due to supercharging must not be allowed



Throttle control is effected either manually or by manifold pressure acting through servomotor. Latter can be locked out when pressure control lever is moved to either extreme position, releasing latch

to exceed safe limits. In normal operation this pressure, acting through a servomotor, operates the throttle to prevent dangerous pressures. When it is desired to render the pressure control inoperative the servomotor is locked in one position so that throttle control can be entirely manual. Lockout means for this purpose are covered by patent 2,310,018, recently assigned to Bendix Aviation Corp. and will be described with reference to this particular application.

Lower part of the illustration shows diagrammatically the carburetor, geared supercharger and intake manifold. By throttling the mixture at the carburetor the super-

charger intake pressure is reduced, effecting a reduction in the manifold pressure. Manual control of the throttle is effected from the quadrant shown in the upper right of the figure through the linkage illustrated, which includes a bellcrank pivoted at its center on the end of the servomotor piston rod.

Manifold Pressure Controls Servomotor

Movement of the servomotor, when released from the locked position, is effected by oil supplied from a pressure source through a piston valve. The porting is such that when the valve is to the right the servomotor piston is urged toward the left, and vice versa. A compression spring also tends to move the piston to the left. Valve position is controlled by the movement of a plate interposed between two flexible bellows of which the left-hand one is evacuated and the right-hand is connected to the intake manifold.

Movement of the plate (and piston valve) is therefore controlled by the manifold pressure and also by the spring shown. Tension in this spring is regulated by movement of the pressure control lever, as indicated in the figure. The greater the spring tension the higher must be the manifold pressure necessary to cause movement of the valve and therefore of the servomotor. Servomotor movement regulates the throttle through the bellcrank, the upper pivot serving as the fulcrum.

Positive Locking Insured

The locking-out of the servo-element, which is the subject of the patent, is accomplished in two ways. First, movement of the pressure control lever to the position shown pulls the plate (and piston valve) to the extreme right. With the valve held in this position the servomotor piston is held in its extreme left position both by oil pressure and by the spring in the servomotor cylinder. In the second place, with the controls in this position the control cam which is included in the pressure control linkage permits the latch to drop into the groove on the servomotor piston rod, preventing further movement. With the center pivot of the bellcrank fixed, the throttle control alone can effect throttle movement.

To release the lockout, the pressure control lever is moved to the left (in the zone *B* to *C*), the plate and piston valve are released, and the cam trips out the latch. When the pressure control lever is moved to its extreme left position (*D*) the cam again allows the latch to drop into the locking position, which it will do as soon as the servomotor piston moves sufficiently far to the left.

SIXTY DAYS ahead of schedule and but fifteen days after the first supercharger was produced at General Electric company's Fort Wayne, Ind., plant, shipment of the first carload of units for the U. S. Army was made. This was six months from the time steel was delivered to this the largest plant of the company's factories, which covers more than 788,000 square feet. Of the total of employes at this plant 58 per cent are women.

ASSETS to a BOOKCASE

Mechanism

By Irving Henry Prageman, associate professor of mechanical engineering, University of Maine; published by the International Textbook Co., Scranton, Pa.; 296 pages, 5¼ by 8¼ inches, flexible binding; available through Machine Design, *$3.00 postpaid.*

For those beginning a study of the motions of machine parts this book will be of practical value. Treatment of the various topics has been made brief so that fundamentals will not be obscured by a large amount of detail, and only those machine parts that are of practical importance are discussed. Numerous examples have been included and the various principles presented are explained with the aid of examples.

Following an introductory chapter covering definitions and a review of fundamentals, the book goes on to discuss linkwork and displacements, velocities in machines, static forces in machines, accelerations in machines, cams, gears and gear teeth, gear proportions and manufacturing methods, gear trains, flexible transmission, intermittent motion, reversing, clutch and brake mechanisms, and diverse types of mechanisms. In an appendix a proof of Coriolis' law is presented and the book ends with a selected list of references.

In connection with this and other recent books on the subject, it is gratifying to note the trend toward an adequate treatment of accelerations in machines. A noteworthy discussion has been included in the book under review showing how acceleration calculations are used to find the inertia loads acting on a connecting rod. Such an example drives home the point that acceleration diagrams have practical use as well as being good exercises.

□ □ □

Industrial Radiology

By Ancel St. John and Herbert R. Isenburger; published by John Wiley & Sons Inc., New York; 298 pages, 5¾ by 9 inches, clothbound; available through Machine Design, *$4.00 postpaid.*

Practical aspects of industrial radiology are here presented in a form and language which can be understood well by engineers. Basic theories are adequately covered without obscuring the essential value of this new technique. While great strides have been made in applications of radiology there are many other potential uses to which this book may serve to draw attention.

Early chapters are devoted to a brief history of radiology, the nature and properties of X-rays, and mutual influences of X-rays and matter. Equipment is discussed in three chapters on production, detection and recording of X-rays, X-ray generators, and description of a typical installation for industrial radiography. Techniques are covered in a series of chapters on making a radiograph, photographic procedure, and interpreting radiographs, in-

dustrial fluoroscopy, radiography of large castings, forg-ings, welded vessels, structures and small objects, also radiography with gamma rays. Concluding chapters are concerned with radiographic specifications and inspection, and operating and cost data. In a series of appendices are presented radiographic rules, tables, charts, and a 1314-item bibliography. The index covers references in the bibliography as well as those in the text.

□ □ □

Tool Design

By Cyril Donaldson and George H. LeCain, Rochester Athanaeum and Mechanics Institute; published by Harper & Brothers, New York; 443 pages, 6 by 9¼ inches, clothbound; available through Machine Design, *$3.75 postpaid.*

Covering general methods of tool design, this book is aimed primarily at those preparing for positions as tool designers. Because of the influence of tooling on machine design, however, the subject is one with which all designers should be thoroughly familiar. The book should therefore appeal to experienced engineers whose work involves consideration of tools, jigs and fixtures, gages, etc., in design of machines, though not necessarily requiring that they design the tools themselves.

The introduction is followed by chapters on drafting room practice, manufacturing processes as they affect the designer, properties of materials, tolerances and allowances, springs, welding, cutting tools, punch and die design, gage design, elementary jigs and fixtures, details of jigs and fixtures, practical design of jigs and fixtures, construction of and tools used on Brown and Sharpe automatic screw machine, cam design, and turret lathes.

Profusely illustrated by photographs and by line drawings of unusual excellence, the book is thoroughly practical and explains principles in terms of actual examples of tool design.

□ □ □

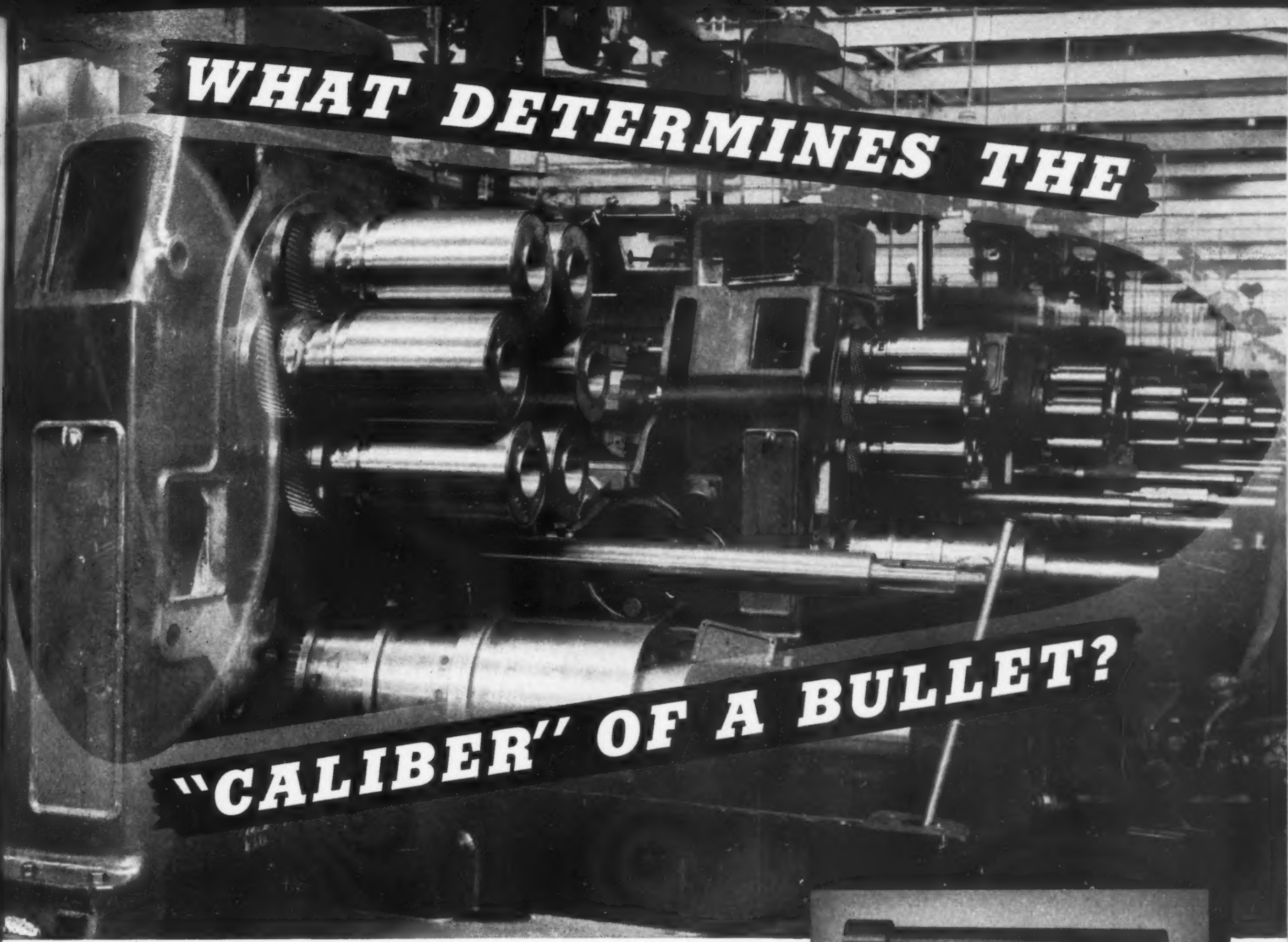
Commodity Year Book

Master edition, published by Commodity Research Bureau Inc., New York; 413 pages, 8½ by 11 inches, clothbound; available through Machine Design, *$7.50 postpaid.*

Covering virtually all the important raw and semifinished products, this new edition has gone considerably further than preceding editions. An encyclopedia of materials, the book lists 836 items, the more important of which are discussed in articles extending to several pages.

Each commodity is analyzed on the basis of a general outline covering such points as physical description, method of production and area of origin, principal uses and finished products, marketing and transportation methods, comparative prices in recent years, perishability, principal types and grades, possible substitutes, government regulations, and import duties. Metals, oils, plastics, chemicals, etc., of industrial importance are covered, as well as foodstuffs, fibers, skins and other items.

WHAT DETERMINES THE "CALIBER" OF A BULLET?



OSTUCO "semi-tailors" seamless tubing parts for precision tools . . . .

(A) Samples of OSTUCO tubing supplied to the National Acme Company; (B) Finished spindles, and pusher tubes used in the Acme-Gridley machines illustrated above. Upsetting for spindles done by OSTUCO. Unusually thin-walled tubing furnished with high tensile strength. Inside of tubes is commercially smooth finish supplied by mill. Forming, and finishing operations at mill cut number of operations this manufacturer must make. (Photos courtesy of National Acme Company.)

VITAL COGS in the arsenal of democracy are the precision machine tools which insure the high caliber of workmanship that goes into every round of ammunition.

Long before Pearl Harbor, OSTUCO supplied machine tool manufacturers with seamless steel tubing, drawn, finished and formed to individual specifications for such parts as spindles, collet and pusher tubes. The ability to meet these rigid specifications substantially reduces machining operations, an advantage that counts even more today when the demand for greater production must be met without any let-up in quality. And while Victory gets first consideration now, OSTUCO is looking ahead to applying newly-acquired skills, techniques and experience to the post-war developments your new experience is certain to produce.

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THE OHIO SEAMLESS TUBE COMPANY

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DESIGN ABSTRACTS

Designing Plastic Parts

THREE major considerations in plastic design are: (1) Selection of the proper material to give the desired physical properties; (2) selection of a means of manufacture which is suitable and economical; and (3) proper part design to enhance the material's qualities and to facilitate manufacture.

For the sake of discussion, let us assume a plastic part is wanted. The part could be made by one of the following methods:

1. Standard sheets, rods and tubes of almost any plastic material in a wide range of sizes could be drawn, machined, punched or otherwise fabricated and assembled into the part. There would be no tool cost, and the design or material could be changed at will. But there would still remain the problem of machining and assembling each unit.

2. A steel master of the part could be made. This mandrel, when dipped into molten lead, will form lead shells which serve as molds. Cast phenolic resin, acrylic resin or styrene resin could be poured into these shells to produce the plastic part. This would cost only one to two hundred dollars for the tools, but the materials are limited and the design is limited to simple shapes and wide tolerances.

3. If the part is of constant, not too heavy cross section, an extrusion die can be made and the part "squirted" in continuous lengths. A simple die is quite inexpensive, but this method is suited only to continuous lengths and principally to thermoplastic compounds.

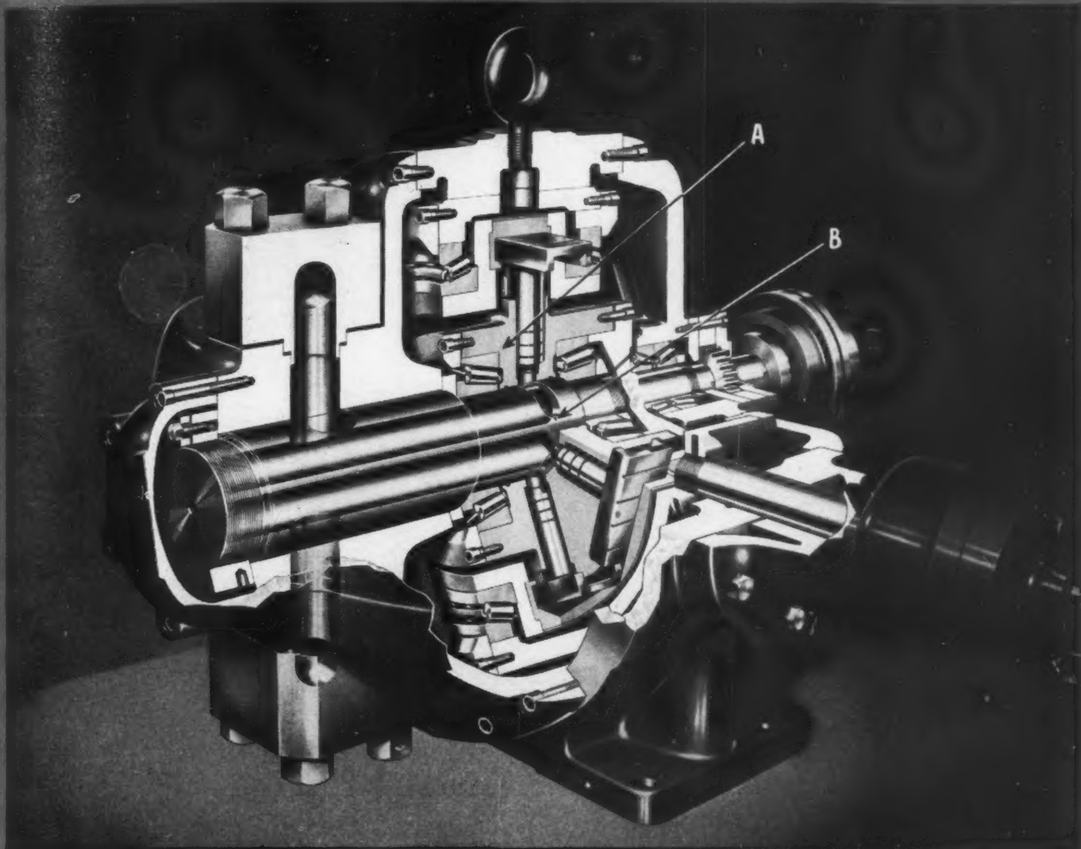
4. If the part is fairly simple in shape and can be made of laminated fabric or veneer, an inexpensive mold can be made and the part produced by low pressure laminating. This is the method being used to produce airplane fuselages and sections from plastic laminates.

5. A hardened steel mold could be made and the part molded exactly to size. The mold would cost about $500 and might run much higher. But this method is advantageous in that molded parts, even if complicated, can be reproduced rapidly, dependably, and as a finished product.

Molding Methods Are Listed

By far the greatest percentage of plastic parts are molded by the last method; we shall recognize it as the most important, and devote our comments on design to it. The use of this hardened steel mold will vary according to the type of molding which we choose to use. We have our choice of the following:

a. Compression molding, in which the plastic compound, usually thermosetting, is molded under heat and


A
B

pressure. A charge of compound is placed in the mold cavity and the plunger, as it enters the cavity, forces the material to the shape of the mold. The mold is heated to 350 degrees Fahr. and the pressure of molding compound will be from two to ten thousand pounds per square inch.

b. Transfer molding, in which the plastic material is charged into a separate well; a plunger entering the well forces the compound through a sprue and runner into the part cavity. Since there is very little pressure in the cavity until it is full, a weak mold section or a delicate insert can be safely used.

c. Injection molding is designed primarily for thermoplastic materials. These materials are heated and kept fluid in a cylinder. During the molding cycle, this material is forced through a nozzle into a cool cavity, where it loses heat and solidifies.

d. Cold molded parts are pressed under pressure alone; the heat is added and the parts are cured in a baking operation which follows the pressing. This method produces a coarser looking part, with broader dimensional tolerances.

e. Molded laminated parts can be pressed from impregnated paper or cloth; the design must be quite simple and the cross section of the part uniform.

How Material Properties Affect Design

It is important to visualize the material properties and the molding operation so that we can properly allow for them in design. For example, if the part has a low impact strength, we must provide thicker walls and fewer projecting fins; if a part is to be compression molded, at high pressures, we must have sturdy inserts and a strong mold.

Fundamental, of course is that we must be able to get the part out of the molds; if undercuts are necessary, it will mean extra mold parts expense. And since our parts must come out, we should have at least .008-inch per inch taper on the vertical walls.

Design to avoid as much machining as possible; tapping, drilling, grinding and machining of undercuts is frequently justified, but machined surfaces are susceptible to wear and to moisture absorption.

Consider the mold which is to be made; avoid sharp interior corners by providing radii on the part. There should be no radius at the cutoff corner, however. Do not require unnecessarily close tolerances. In general, on a fixed mold dimension, tolerances can be held to plus or minus .003-inch per inch. On dimensions which are affected by the closing of the mold, an additional .01 to .02-inch is required. You should allow about .003-inch for warpage per linear inch of flat surface.

For phenolic parts, an average wall thickness is 3/32-inch, and the minimum should be .025. Injection molded thermoplastics are preferably .07 to .125-inch thick. Keep the wall thicknesses uniform, with no abrupt changes in section. Plastic parts shrink as they cool, and the uneven sections cause strain and cracking.

Brass, steel, aluminum—inserts of almost any type—can be molded into plastic parts. They should be uniformly made to fit the mold, and designed to resist the molding pressures and temperatures. Grooves, knurls or other irregular surfaces must be provided to anchor the

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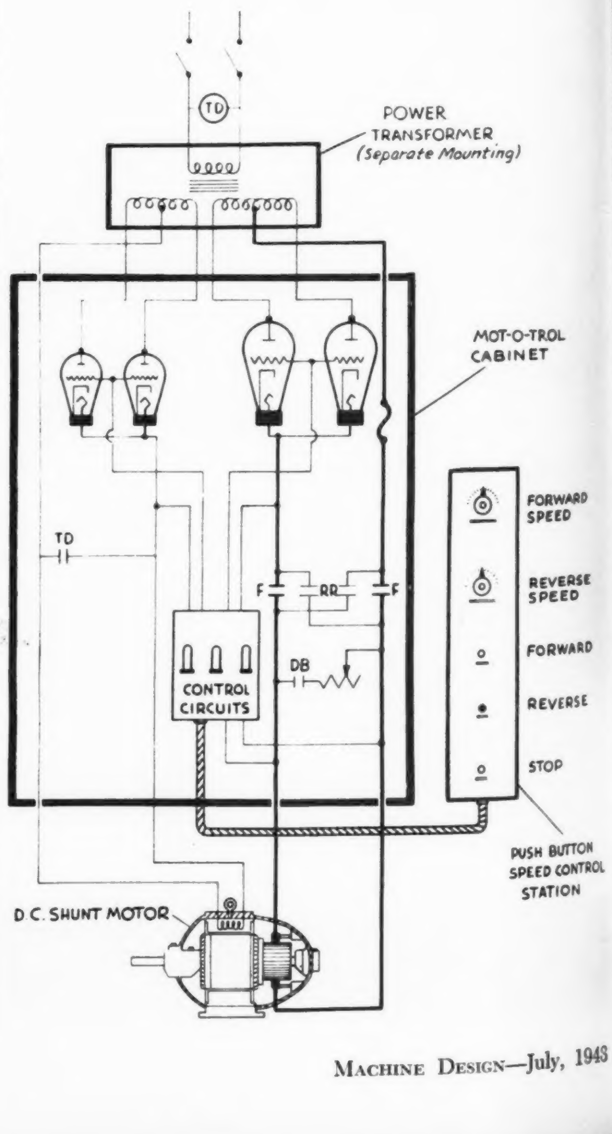
insert to the compound.

Remember that for every hole or depression, there must be a corresponding mold part; if it is too weak or too slender, it will break or bend in molding. Molded threads larger than a number 8 with less than 32 threads per inch are quite simple to mold, and are very satisfactory.

Good part design involves just the exercise of good judgment and a familiarity with the material used and the process of manufacture.—*From a recent paper by W. S. Larson, General Electric Co.*

Electronic Motor Speed Control

RECENT refinements in alternating-current variable-speed drives with electronic control eliminate many of the earlier handicaps and make the new systems comparable to or better than other existing solutions. In general the system, as shown in the illustration, consists of a single or polyphase grid-controlled, thyratron tube rectifier which takes power from an alternating-current line and rectifies it into direct-current output. The rectified direct-current voltage is applied to a regular shunt-wound direct-current motor and may be varied from zero to motor rated voltage (or above) for direct-current armature control. Smaller thyratrons also are used in the control to provide rectified direct-current field current for the



motor, the field voltage being held constant throughout the range of armature voltage, then reduced to provide greater speed range by field weakening.

The diagrammatic sketch represents a Mot-O-Trol system for one horsepower and smaller, using single phase full-wave rectification on both field and armature. Four pieces of equipment are involved: The small power transformer, for separate mounting on the original machine, the Mot-O-Trol cabinet proper, control station, and direct-current motor.

Speeds may be preset to any desired speed within the design range with two speed-control potentiometers and reversing contactors as indicated. Different forward and reverse speeds may be preset so that only the operation of the forward or reverse pushbuttons is necessary to obtain a predetermined speed in either direction. Speed also may be adjusted at any time while the motor is running. Speed control potentiometers are tandem type to cover the entire range of armature and field adjustment on a single dial. Adjustment changes the firing point on the alternating-current line voltage wave at which the tubes fire and thereby varies the output direct-current voltage as required.

Extensive Speed Range Available

Normal speed range by armature control is 20 to 1 (below the base speed of the motor) though a wider range such as 100 to 1 is obtainable. Above basic speeds field control is used, normally 2 to 1 for standard motors but limited chiefly by the mechanical limitations of the motor.

The standard drive is designed to regulate automatically the motor speed, if adjusted for any one speed, so as to maintain essentially constant speed regardless of load. Through other small, inexpensive control tubes the direct-current voltage output of the main rectifier tubes is controlled to compensate for speed changes. The degree of compensation can be adjusted from within the control cabinet. In a properly adjusted system the speed will not vary more than 4 per cent from a presetting (with torque varying from no load to full-load value) for a speed range as much as 10 to 1, nor more than 8 per cent for any speed within the range of 20 to 1. Normal variations in alternating-current line voltage have only a small effect on the speed regulation.

Motors furnished with the Mot-O-Trol system are selected to handle constant torque load over the entire armature control speed range (or up to the base speed of the motor) and constant horsepower over the field control range, continuously without exceeding safe temperature limits. The frame size of the motor will depend upon the base speed rating and other operating characteristics but in most ratings will be somewhat larger than the standard direct-current motor frame of the same horsepower and basic speed rating. The reason for a larger motor frame in most cases is due to the high form factor of pulsating current obtained from the rectifier when the firing angle is phased back to obtain low output voltage and consequently low motor speeds—*From a paper by T. R. Lawson, electronic control engineer, Westinghouse Elec. & Mfg. Co., presented at the recent Machine Tool Forum.*

High-Frequency Heating Speeds Plastic Molding

(*Continued from Page* 103)

rial together form a condenser. If the material were a perfect dielectric no heating would occur. However, all present-day thermosetting materials are sufficiently far from perfect dielectrics that the highly oscillating field generates enough heat in the material to raise it to molding temperature within a matter of fifteen to sixty seconds.

It may be worth while to show the basic formula for heating by this means:

$$H = Kf\ (L.F.)\ (E/t)^2$$

where H = heat per unit volume, K = constant, f = frequency, $L.F.$ = loss factor, E = voltage, and t = thickness of sample.

It will be noted that heating varies directly with frequency and loss factor, and as the second power of the term E/t, the potential gradient across the sample. For a given material the loss factor is fixed. As a general statement, the higher the frequency of the current, the more it costs. It is therefore logical to go as high as possible on the E/t term. The limitation in going up in potential is the danger of dielectric breakdown or arcing between the electrostatic plates. This limitation makes it desirable to work at frequencies of ten megacycles or higher, which is quite feasible today since power tubes are available within this range, although obtaining them requires a convincing argument to the War Production Board. As one attempts to go to tubes operative at considerably higher frequencies, difficulties in operation and procurement multiply.

Automatic Operation Desirable

The present simplest procedure applied to thermosetting plastics consists of heating the material to or near its molding temperature, quickly transferring the hot material to the mold and closing the mold to complete plastic flow prior to any substantial hardening of the material. The time available for this operation will vary somewhat for different materials. High temperatures are an advantage in speeding up the molding but the process also becomes more critical and therefore the best available compromise has to be made for each set of operating conditions. It will readily be appreciated that automatic operation of all sequences is most desirable if the process is to be speeded to its ultimate.

As will readily be realized, the rate of heating must bear a relationship to the speed at which the plastic hardens. Thus if heating requires too long the material will commence to harden before flow is finished in the die. For the present general-purpose phenolic it is desirable to have a machine of about five kva input per pound of material. If 50 per cent conversion of electrical energy to heat is obtained, as may be expected, this means 2.5 kva per pound, which should heat the material to molding temperature in forty-five seconds or less.

It is advisable to have the molding material in a preformed state if best uniformity and heating efficiency are to be obtained. While powder will heat, it will be at

a lower rate than with preforms and there is more danger of differences in temperature developing within the material being heated. Although it is not necessary to have absolute uniformity in order to show a distinct advantage over present compression molding, as the process becomes more automatic and is speeded up to its ultimate, the uniformity of temperature will assume great importance. Factors influencing uniformity are:

1. Uniform density of material, generally the highest density being preferred
2. Uniform thickness and shape of preform.

A point not usually appreciated by those unfamiliar with the technique is that uniformity is greatly influenced by the shape of the part to be heated. Although odd shapes can be heated it is simpler if uniformly thick flat preforms can be used.

Cure Time Not Dependent on Heat Transfer

If the best principles of this new process are followed, certain definite and distinct advantages are obtained. The first of these is shown in *Fig.* 1 from which it will be noted that cure time is almost independent of thickness of section. This great difference is due to the fact that the material is uniformly heated prior to molding and therefore the time in the mold is not greatly prolonged by the slow rate of heat transfer.

A second advantage is increased plasticity. Data are presented in the form of "closing time" of a standard cup mold, as shown in *Fig.* 2. That is, the time to close the die under definite conditions of temperature and pressure is a practical measure of the plastic properties of the material. The shorter the time the more fluid the plastic.

Work done is in all cases substantially less using Heatronic molding. However, the work is done in so much shorter time that the power, which is rate of work, is usually higher. This simply means in most cases that lines from the accumulator will have to be larger. In other cases, such as self-contained units, it may mean lower pressure, higher volume pumps. Phenolic thermosetting materials have thus far been discussed, with occasional references to the urea type. The phenolics appear unusually well adapted to this process. However, other varieties such as urea, melamine, alkyds and various types of rubber can be heated quite readily and reductions both in molding pressure and time of molding are obtained. It is therefore anticipated that general use will be found for the process.

Avoids Prolonged Heating of Thermoplastics

Advantages in the thermoplastics field are not as striking as for thermosetting, which is not to say that substantial uses will not be found since in some respects thermoplastics also are limited by heat transfer. For instance, in the molding of large parts, recourse has been had to multiple cylinders. In the handling of heat-sensitive products various forms of torpedo construction and cylinder streamlining have been employed. All these are really different manifestations of the tendency of all thermoplastics to break down if subjected to high temperatures for long periods of time. Heatronic heating will raise the material to molding temperature in a relatively short period

The Heil Company of Milwaukee, Wis., has furnished the Army and Navy with numbers of these refueling units. Pumps, clutches and control valves are in cabinet on side of truck. Above is view of pump shaft, with Link-Belt Silent Chain driving from power take-off.

● Efficient, dependable pumping of gasoline or oil from these "dry-land tankers" to Navy planes or boats demands a power drive that is reliable under the most severe conditions. Link-Belt *Silverstreak Silent Chain was selected for this vital function, an outstanding recognition of its performance advantages and inherent stamina. Silverstreak Silent Chain Drives are well fitted to meet such exacting requirements—temperature and atmospheric conditions have no effect on its smooth, trouble-free operation. Power is transmitted without slipping, delivering every r.p.m. without fail. It requires no attention beyond occasional lubrication. Under severe conditions in numerous diversified industries, Link-Belt Silverstreak Silent Chains have made records of 10—15—20 years of continuous operation at amazingly low cost figures. Send for Data Book 125.

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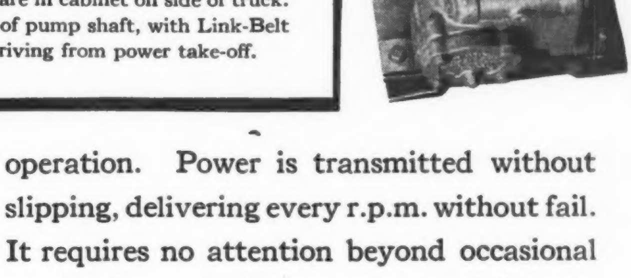

of time, thus removing the necessity for multiple heating cylinders and for long periods of time in the heating cylinder. A great advantage not only in making large parts but in minimizing plastic breakdown has therefore been gained.

Savings in Power Requirements Effected

Another point also should be stressed. The pressure exerted by the injection ram in present injection molding is often absorbed to the major extent in attempting to move cold material through the heating cylinder. Pressure dissipated in this way is not effective for producing positive effect on the molded piece in the die. If plastic material is charged to the cylinder, this source of pressure loss will be removed, resulting in less power requirement to do a better job than at present.

In conclusion the following advantages are characteristic of the process for molding thermosetting materials:

1. Increased speed of hardening, especially for thick sections
2. Increased plasticity which enables greater production from the same press and die or a smaller press and die for the same production
3. Less wear on dies
4. Less danger of pin and insert breakage or displacement
5. Probable use of lower cost dies of cast or plate types where only limited production is desired and cost is a considerable factor, due to the lower pressures and less abrasive nature of the plastic
6. Feasibility of molding large parts on a fast cycle.

For thermoplastics, advantages are to be found in:

1. Elimination of the present heat transfer problem in heating cylinder design and thereby allowing for large moldings from a single heating cylinder
2. Lower power requirements for the same moldings
3. Less polymer breakdown giving molding parts of improved quality
4. Mitigation of present stringent requirements for heat resistance

While no particular mention has been made of the strength of molded parts, exhaustive data by well controlled comparative tests have shown molded specimens equal to or better than those obtained by present conventional methods.

Vibration Isolation in War Machines

(*Continued from Page* 129)

curate knowledge of weight supported; an improper selection will almost inevitably result in poor isolation characteristics.

In a resilient mounting installation without such snubbing, a large displacement can result in metal-to-metal contact of bracket members. Accordingly, it is usually desirable to set limits to travel, either early or late in the stroke, employing a stiff, though far from rigid, medium. Recommended travel in either direction before undergoing snubbing action should be seven to ten times the single amplitude of the most destructive high frequency compo-

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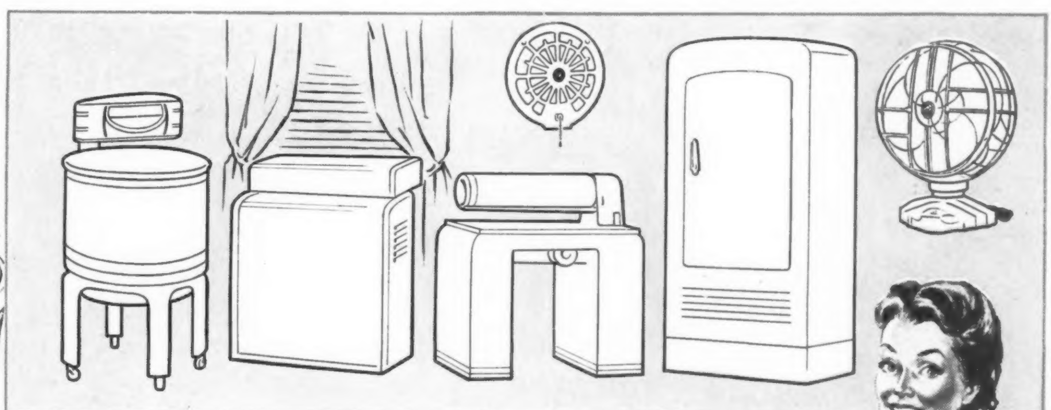
THE ELECTRIC MOTOR ENTERS THE HOME!
Sewing Machine reproduced from Emerson-Electric advertising of 1899



TYPES OF HOME APPLIANCES POWERED BY EMERSON-ELECTRIC MOTORS

"Until the war, I never fully appreciated the importance of the electric motors on my home appliances."

When war came, the entire resources of Emerson-Electric's 53 years' experience were quickly converted and tremendously expanded for manufacturing vital implements of war—power-operated revolving gun turrets, shell parts, and many new types of electric motors for aircraft.

Out of the urgencies of war will come entirely new conceptions of electric motor design, construction and efficiency. "After Victory", manufacturers of the new and improved motor-driven appliances and equipment will confidently power their products with these motors.

In recognition of their "patriotism and great work", Emerson-Electric workers were presented with the Army-Navy "E" Award.

THE EMERSON ELECTRIC MANUFACTURING COMPANY
SAINT LOUIS • *Branches:* New York • Detroit • Chicago • Los Angeles • Davenport

EMERSON EMERSON ELECTRIC ELECTRIC

MOTORS • FANS • APPLIANCES • A. C. ARC WELDERS

nent of disturbance; at the same time this clearance should be at least equal to the amplitude of the lower frequency component. If it is found to be entirely impractical to provide an assembly according to this plan, because of space limitations (it is to be borne in mind that the safe travel for a rubber mounting is proportional to the rubber dimension being deflected), probably the worst alternative would be to incorporate snubbing at some shorter degree of travel. Such an arbitrary choice has been known to result in the magnification of vibratory forces due to the effect of blocking the movement of the system when its velocity is high. It is felt that a decidedly better compromise than this is to incorporate earlier snubbing action (which does not stiffen up sharply at any stage of deflection), in order to check the movement before high velocity can develop.

Consider Center of Gravity Location

In approaching a conclusion on this subject, a number of practical points can be mentioned which have been found to be of advantage. Particularly important is the advisability of locating mountings in either a horizontal or a vertical plane with the center of gravity of the supported equipment. The advantages of a "center of gravity suspension" are: (a) It substantially prevents different modes of vibration from becoming coupled; (b) the individual natural frequencies can be made to fall within a reasonably narrow range; and (c) for a given stiffness of mountings this arrangement yields the greatest possible stability.

There is no question that rubber mounting engineers would be in a position to extend greater service in devising outstandingly protected equipment installations if the attitude "Here's a piece of apparatus; how will we mount it?" were superseded by: "We are laying out a new device. What provision shall we make for rubber mountings?" Nevertheless, in the necessary haste of our present problems, we are often compelled to do the best we can with what exists now. This fact has often been the reason for locating the points of resilient support at the base of a tall piece of equipment as in *Fig. 8a;* the stability of this assembly, with center of gravity high above the mounting plane, would be greatly improved by the addition of stabilizing mountings at the top as indicated in *Fig. 8b.*

Providing More Protection for Individual Elements

Sometimes an individual delicate element in an apparatus requires a greater degree of protection than can be provided within the overall limitations of a resiliently mounted installation. This might be true of a particular radio tube in a set mounted at the cabinet base. An arrangement that has been applied to provide the required protection in such instances is the separate mounting of that single element as a unit. Having done this, one is not to expect this secondary mounting system to exhibit a natural frequency that can be anticipated solely in terms of its own static deflection, since the resilient supports at the base will also have a bearing on the combined natural frequency. Before freezing such a design, it is recommended that the complete apparatus be subjected to shake-table tests and that all mounting selections be fixed on the basis of the observations made.


"Tin Fish"..
Globe Steel Tubes Help to Make This Deadly Weapon of Modern War

Aspects of Plywood

(*Concluded from Page* 119)

tive magnitudes of the net horizontal and vertical forces acting on the gusset.

In general, it can be stated that the plywood gusset should be 50-50 construction of not less than three plies. For ease in gluing, the density of the outer plies of the gusset and the density of the cap-strip and diagonal material should be approximately the same.

If the greatest strength is desired in the vertical direction and the glue area between the gusset and cap-strip is critical, the plywood face-grain direction should be run parallel to the rib cap-strip, assuming 50-50 construction is being used. If the greatest strength is desired in horizontal shear, such as when the horizontal forces in the diagonals are additive, the plywood gusset should be laid so that its face grain makes an angle of 45 degrees with the cap-strip. A further point to be noted in the latter case is that the plywood face grain should run in the general direction of the tension load, *Fig.* 6.

STRESS CONCENTRATIONS: The proportional limit of wood in tension is very near its ultimate strength. This, combined with the weakness of wood in shear and cleavage makes it essential that abrupt changes in section, re-entrant cuts, odd-shaped lightening holes, and the like be avoided wherever possible. Unfortunately, stress concentrations in wood cannot relieve themselves by plastic yielding as do metals.

Reinforcing blocks should be tapered in thickness or in width or both, so that they will pick up stress as uniformly as possible. When fish-mouthed filler blocks are used in box-spars for stress transfer, the taper ratio of the fishmouth should be determined by the ratio of the maximum tension stress existing in the adjacent flange of the horizontal shear strength of the filler block, *Fig.* 4.

REPLACING SOLID GLUE BLOCKS WITH PLYWOOD ANGLES: A few tests which have been made indicate that plywood angles (bent up from 45 degrees plywood) will carry safely 250 pounds per square inch in shear. Since their weight is only one-third to one-half that of an equivalent solid glue block, some weight saving can be realized.

REDUCTION IN FITTINGS: A staggered bolt pattern which will give a smaller and more efficient fitting can be used if suitable plywood plates are glued to the faces of the bolted members. The weight of a wood airplane is greatly increased by the use of extra material required near fittings; consequently these should be held to a practical minimum in any design.

The somewhat poor reputation that wood aircraft structures have acquired in the past year in static tests can be attributed to a lack of knowledge of plywood strength and elastic properties, as well as to poor detail design practices. It should be remembered, however, that both of these items will be overcome as a background of design information and experience is gained similar to that already available for metal structures. What constitutes poor detail design practice in any type of structure may not be realized until after a costly static-test failure or service failure is encountered. Such failures provide the foundation for more efficient designs in the future.

BUSINESS AND SALES BRIEFS

DUE to the death of Carl M. Peterson, who had been secretary and treasurer of Star Electric Motor Co. and its affiliate the Star Equipment Corp., a new executive staff has been appointed. Elvin E. Hollander was made first vice president of both concerns, Ivor C. Peterson vice president in charge of sales, and R. J. Gash secretary and treasurer.

♦

Meehanite Metal Corp., New Rochelle, N. Y., has announced that the John Hastie & Co. Ltd., Greenock, England, is now producing Meehanite castings, chiefly for hydraulic steering gears.

♦

Succeeding his brother, Maurice C. Bachner, who becomes board chairman, Edward F. Bachner has been elected president of Chicago Molded Products Co. He formerly occupied the position of general manager.

♦

Award of the John Price Wetherill medal to Robert H. Leach, vice president, Handy & Harman, Bridgeport, Conn., has been made by the Franklin Institute.

♦

Formerly sales manager, William R. Waddell has been made manager of the service division of Federal-Mogul Co. Don Switzer, who had been assistant sales manager, has been named sales manager.

♦

Herbert M. Rich Jr. has been appointed district manager in New York for the Electro Metallurgical Co., a unit of Union Carbide & Carbon Corp.

♦

Palmer-Shile Co., Detroit, has been placed in general charge of all sales of Briggs & Stratton Air-Saver air valves. The equipment will continue to be manufactured and marketed as a Briggs product.

♦

Appointment of William W. Klemme as district manager of industrial sales in Dallas, Texas, has been made by Chain Belt Co.

♦

Associated with Dow Chemical Co. for the last twenty years, D. L. Gibb has been appointed manager of the plastic sales division.

♦

C. W. Higbee, manager of wire sales for United States Rubber Co., New York, has also been made manager of the wire and cable department. The new manager for aeronautical wires and cables is E. S. McConnell.

♦

To speed the production of special type radio equipment for the Army, Westinghouse Electric & Mfg. company's radio division has announced the opening of a new four-acre blackout plant on the east coast.

♦

Formerly head of Elastic Stop Nut Corp. of America, E. A.

Aircraft: Lyon Aircraft Division is fabricating aluminum and steel parts and subassemblies for aircraft being delivered by prime contractors to both the Army and Navy.

Sheet Steel: Forty-three years' experience in fabricating steel—nearly three years on special war products.

Sheet Metal Stampings: Facilities for handling a wide range of gauges, sizes and drawing operations including annealing. Experience on conversion of castings and forgings to sheet metal.

Ships: Experienced Lyon Engineers are working daily with architects and shipbuilders on prefabricated parts and furniture.

Know How: Lyon war contract experience dates from July, 1940. Every inquiry is given "specialist" handling. Experienced development, design and engineering staffs. Complete toolrooms in all plants. Send for this new book, "Craftsmen in War Production." Describes facilities for aluminum and steel production.

LYON METAL PRODUCTS, INCORPORATED

General Offices: 8107 Madison Avenue, Aurora, Illinois

Sales and District Offices Manned by Experienced Engineers in All Principal Cities

LYON METAL PRODUCTS, INCORPORATED

Swanstrom has been appointed president and general manager of the newly formed Penn Engineering & Mfg. Corp., Doylestown, Pa., organized to manufacture self-locking nuts and aircraft fittings. B. C. Sandemar is vice president and chief engineer of the new organization.

♦

Recenty associated with Thomas Mason Co., George W. De Bell has opened offices in the Gurley building, Stamford, Conn., where he will practice as a plastics consultant.

♦

Removal of its stock products sales office from New York city to the plant at Buffalo has been made by J. H. Williams & Co., manufacturer of drop forgings. A local sales office will be maintained, however, in New York to serve the metropolitan area.

♦

An addition is being made to the plant of Moore Drop Forging Co., Springfield, Mass., for the manufacture of aircraft parts.

♦

With headquarters at 163 Linden avenue, Oak Park, Ill., L. A. Shea will act as district manager for the Chicago territory of Henry L. Crowley & Co. He previously was associated with Moraine Products division of General Motors Corp.

♦

Opening of a plant in Burbank, Calif., has been announced by Aircraft Accessories Corp. This is plant No. 3 and is under the supervision of W. A. Ashton.

♦

Max Pischke has assumed the duties of manager, with the title of acting district manager, of Allegheny Ludlum Steel Corporation's Pittsburgh district sales office. He replaces Robert H. Gibb who has accepted a commission in the United States Navy.

♦

Greene, Tweed & Co., New York, manufacturers of packings, couplings, etc., has announced the appointments of J. J. McIntosh as southeastern sales representative and B. F. Coombs as Texas sales representative.

♦

In the state of Michigan, American Equipment Co., 5928 Second boulevard, Detroit, will act as representative for the American Engineering Co., Philadelphia, in the sale of its Hele-Shaw pumps.

♦

After thirty years' service with Carpenter Steel Co., Reading, Pa., Richard Calvert has retired as sales representative in eastern and central Pennsylvania. Avard Taylor will continue for the company in this territory.

♦

Succeeding Gordon F. Hess, E. E. Haubegger has been appointed district sales manager for Republic Steel Corp. in Houston, Tex. Mr. Hess was recently made district sales manager for Republic in Detroit.

♦

Previously sales engineer for the Bohn Aluminum and Brass Corp. and the Dow Chemical Co., G. C. Brown has been appointed Detroit engineering representative of the Aircraft Screw Products Co. Inc.

♦

To accommodate its increased activities, Shafer Bearing Corp., Chicago, has purchased a three-story building at 1412 West Washington boulevard.

♦

Controlling interest in the stock of Electroweld Steel Corp.,

Oil City, Pa., was recently purchased by the Talon Inc., Meadville, Pa.

♦

Succeeding L. M. Hogan, resigned, I. H. Anderson has been named district manager of sales for the Steel and Tubes division, Republic Steel Corp., New York.

♦

J. A. Ireland, division sales manager, has been made assistant general manager of sales of the Steel and Hubes division, Republic Steel Corp., Cleveland.

♦

Stackpole Carbon Co., St. Marys, Penna., has appointed Henry Dressel as its supervisor of electronic components engineering. Mr. Dressel has been a member of the company's engineering staff for several years.

♦

Formerly general manager of Taylor-Winfield Corp., John D. Gordon has announced the formation of Detroit Electronic Laboratory with headquarters at 10345 Linwood avenue, Detroit. The new company, of which Mr. Gordon is general manager, is concentrating on the development and manufacture of special purpose electronic tubes.

♦

Quadrupling of its production of Plexiglas sheets for bomber noses, gun turrets, astro domes, and other transparent plane enclosures will result from the opening of a new plastics plant by Rohm & Haas Co., Philadelphia. The new plant is located in Knoxville, Tenn.

♦

For the past five years chief engineer and director of sales of Universal Gear Corp., J. Y. Dahlstrand has been named vice president of the organization.

MEETINGS AND EXPOSITIONS

June 28-July 2—

American Society for Testing Materials. Annual meeting to be held at the William Penn hotel, Pittsburgh. R. E. Hess, 260 South Bend street, Philadelphia, is assistant secretary.

Set. 23-24—

Society of Automotive Engineers Inc. National tractor meeting to be held at the Schroeder hotel, Milwaukee. John A. C. Warner, 29 West Thirty-ninth street, New York, is secretary and general manager.

Sept. 28-30—

Association of Iron and Steel Engineers. Annual convention to be held at Hotel William Penn, Pittsburgh. Brent Wiley, Empire building, Pittsburgh, is managing director.

Sept. 30-Oct. 2—

Society of Automotive Engineers Inc. National aircraft engineering and production meeting to be held at the Biltmore hotel, Los Angeles. John A. C. Warner, 29 West Thirty-ninth street, New York, is secretary and general manager.

Oct. 18-23—

American Welding Society. Twenty-fourth annual meeting to be held at Hotel Morrison, Chicago. M. M. Kelly, 33 West Thirty-ninth street, New York, is secretary.